MODERN MECHANICAL ENGINEERING

MODERN MECHANICAL ENGINEERING

A PRACTICAL TREATISE
WRITTEN BY SPECIALISTS

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VOLUME IV

THE GRESHAM PUBLISHING COMPANY LTD.
66 Chandos Street, Covent Garden, London
1923

Printed and bound in Great Britain

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MECHANISM

BY

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Mechanism

In General.—A machine consists of an assembly of parts grouped together to serve a special purpose and usually inter-related in such a manner that there is only one degree of relative freedom between the parts. These parts are either rigid, which have contacts of the higher or lower type (Vol. II, p. 90); elastic, as springs; flexible, as belts; and sometimes fluid. One part in a machine is usually regarded as fixed and is termed the frame. If the relationship of the parts is such that the position of a part is definitely controlled by the position of the driving member of the machine, the part is said to be positively driven, distinguishing it from cases, such as those including a belt or hydraulic transmission, in which the relative positions of the parts may change owing to slip or leakage.

The primary conception of a machine for any purpose is geometrical, i.e. the parts must go through the sequence of constrained movements required. In pure mechanism this is the view taken, the action between the individual parts and simple, or otherwise important, combinations being geometrically analysed. The next view is that which regards it as transmitting energy (by means of forces and stresses) involving the use of suitable resistant material, the mass of which may demand dynamical consideration. The machine is then to be regarded from the point of view of use, and provision made for suitable lubrication and for the adjustment (in necessary cases) for the wear which is expected. Beyond these there are further aspects, such as the convenience and general suitability of the machine, both for its special purpose and for its manufacture.

The three lower pairs, involving continuous surface contact, are suited for the transmission of large forces: they are comparatively easy to produce accurately, and wear is readily detected and usually easily compensated for. Hence they are of high importance in mechanism.

Lubrication.—While bearings have been lubricated from time immemorial, the first critical study of the subject was the experimental work carried out by Mr. Beauchamp Tower, under the auspices of the Institution of Mechanical Engineers (*Proceedings*, 1883, 1885), and the analysis of this work by Professor Osborne Reynolds. The main actions in lubrication were then elucidated, but experimental work continues on some aspects of the problem.

Surfaces working together must not seize and should run with the least

possible friction, which turns energy into heat, mechanically wasting it. The lubricating oil becomes more fluid with rise of temperature, lowering the friction but becoming more liable to be squeezed out by the pressure. Thus oil must be selected to have suitable properties at the running temperature, and the rise in temperature is often considerable when the speed is high.

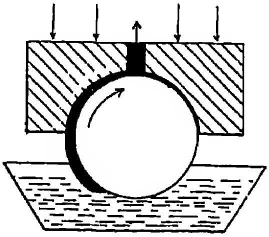


Fig. 1

The "ordinary" laws of friction were deduced by Morin from his experiments on sliding surfaces, under conditions of small velocities and low pressures, and the results obtained were the frictional force F was proportional to the normal force R, and independent of the velocity of sliding and of the area of contact, so that $F = \mu R$, the coefficient of friction μ being about 0·15 for metal on metal, dry or smeared with a "contamination" film only, and 0·075 when the surfaces were "well lubricated". In Mr. Tower's experiments, made on a journal 4 in. diameter × 6 in. long, with a loaded gun-metal bearing on the upper part of the journal, loaded and speeded as in railway practice, and lubricated in some experiments with an oil bath and in others by an oiled pad or siphon, entirely different laws with much lower coefficients of friction were found. The results were most consistent with the perfect lubrication of the oil bath, and it was found that a con-

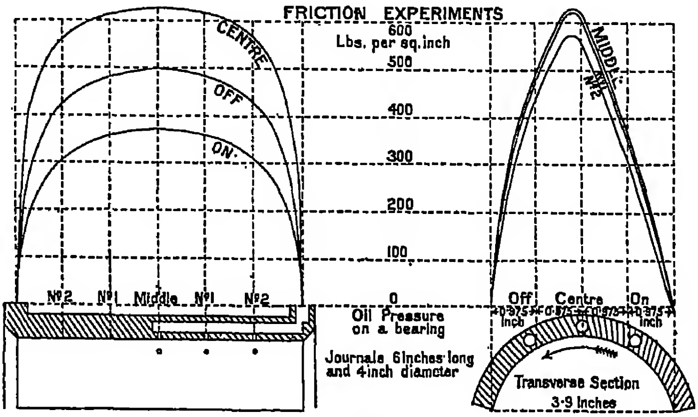


Fig. 2

tinuous film of lubricant was then carried by the journal into the space between it and the bearing, a high pressure being generated in the film. The oil travelled through the small space between the journal and bearing and was delivered on the other side. For lubrication purposes a certain small difference of diameter of journal and bearing is necessary, and in action the axes of journal and bearing are not quite coincident but somewhat as sketched in fig. 1, the oil film varying in thickness.

Oil-film Pressure.—The pressure in the film at various points was measured and was found to be as shown in fig. 2 (*Proc. Inst. Mech. E.*, 1883).

When the oil film is perfect and continuous, the surfaces of journal and bearing are completely separated by it, the action consisting of a viscous shear in the film so that the materials of which the parts are composed are then of no importance. If the load be continuously increased the pressure ultimately ruptures the film and later squeezes out the oil until the surfaces of journal and bearing come into contact, and if the pressure be high enough the bearing seizes, the materials becoming united by a kind of cold welding. To lessen the risk of seizing, with its consequent damage, the materials of journal and bearing should be different—with the exception of cast iron, which works well on cast iron under low pressures. The pairs of metals for journal and bearing are usually considered to be in the following order of merit:

Hardened steel and hardened steel.
Hardened steel and bronze (preferably phosphor bronze).
Mild steel and white metal.
Mild steel and brass.
Cast iron and cast iron.
Mild steel and bronze.
Mild steel and mild steel.

As seizing depends upon squeezing out the lubricant, the more viscous it is the less the risk, and this can be further lessened by the use of solid lubricants (graphite, &c.), but the friction of these is so high as to prevent their general use. Besides viscosity a certain "oiliness" is necessary in lubricating oils: it appears to be dependant on the presence of a small amount of fatty acid. For that reason mineral oils are blended with those of an animal or vegetable origin to produce satisfactory lubricants.

When the oil film is perfect the friction is very low compared with any of Morin's values, and it is nearly independent of the total load and hence varies inversely as the pressure. Some of Mr. Tower's results showing the variation of friction with load and with speed are given in the following table:

Table VI of Mr. Tower's Paper in Proc. Inst. Mech. Eng., 1883, p. 146

(Bath of mineral oil. Temperature 90° F. Medium oil fluid at 50° F. Journal 4 in. diameter × 6 in. long. Chord of arc of contact of mass = 3·92 in. Actual load = 4 × 6 × Normal load).

Normal Load (Lb. per Sq. In.)	R.P.M., Ft. per Min., 100. 105.	200. 209.	250. 262.	300. 314.	350. 366.	410. 419.
625	—	0·0013	0·00139	0·00147	0·00157	—
520	—	0·00113	0·00139	0·00150	0·00161	0·00178
415	—	0·00133	0·00143	0·00160	0·00176	0·00200
310	—	0·00142	0·00160	0·00184	0·00207	0·00241
205	0·00178	0·00205	0·00235	0·00269	0·00298	0·00350
100	0·00334	0·00415	0·00494	0·00557	0·00620	0·00730

The oil film is of the order of a thousandth of an inch in thickness, and is an important factor in the fixing of limits for running fits; the thickness can decrease until films of single molecule thickness—corresponding nearly to Morin's dry metals—are reached. These ultimate films are difficult to remove unless abrasion, as in brake blocks, occurs.

There appears to be no actual discontinuity between starting and running friction, the coefficient, after becoming slightly higher at a low velocity than initially, falls as the speed increases and the liquid film establishes itself, and after a certain velocity the friction varies as the square root of the velocity. When the oil film is broken the action becomes irregular, the friction lying between oil-film values and those found by Morin.

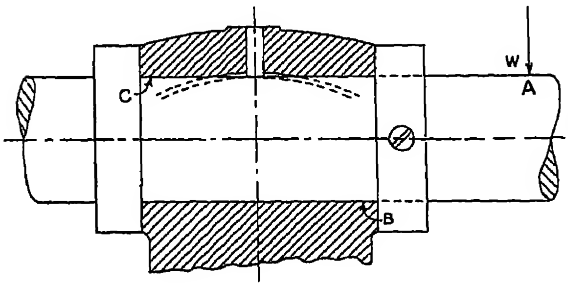


Fig. 3

Oil Holes and Ways.—To supply oil to a bearing it must be introduced at a point of low pressure, or be supplied at a higher pressure than exists in the film at the point of introduction. Thus oil holes and ways must be carefully arranged, or they may be useless. In some of the arrangements experimented with by Mr. Tower he found it impossible to introduce the oil. The ways should carry the supply across the bearing, but must not run to the edge (fig. 3). Entrance of grit with the oil must be prevented, and

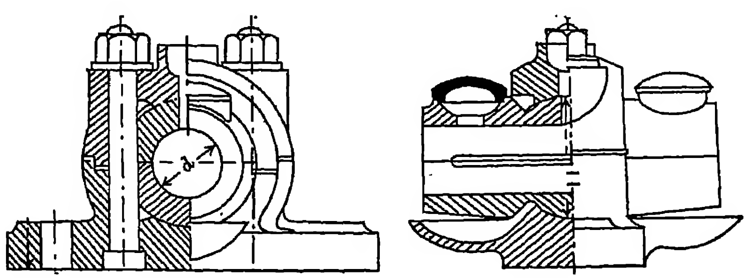


Fig. 4

lubricators (needle type are useful when the shaft runs occasionally) or oil-hole plugs should be used. Common systems of conveying oil to a bearing are by means of wicks and pads (which filter the oil in the action), and by rings or chains dipping into the oil in a reservoir and carried round by the shaft upon which they rest. Worsted wicks cannot be relied upon to raise oil more than 1 or $1\frac{1}{4}$ in., but fine cotton wicks will raise it much farther.

Journal Bearings.—If a shaft be supported by a single bearing, as in fig. 3, the load W (shown vertical) at A makes the journal bear hard

at B and C, leading to wear and possible seizure, and hence a shaft is usually supported by bearings having a clear space between them. Two such bearings must be coaxial, which can be secured by careful fitting or by giving the bearings freedom to adjust themselves slightly. Such a bearing, arranged as a plummer block, is shown in fig. 4, and is known as the Sellars' type. They have the added advantage of adjusting themselves to any slight bend of the shaft under its load. Such bearings should be employed for line shafting, and are then fitted with horizontal and vertical adjustments so that the shaft can be lined up accurately: line shaft bearings should be of the split type for replacement purposes. In fig. 5 is shown an arrangement for oiling the crank pin which is largely employed for slow-speed engines, but in modern high-speed engines forced lubrication, or at least splash, is coming increasingly into use.

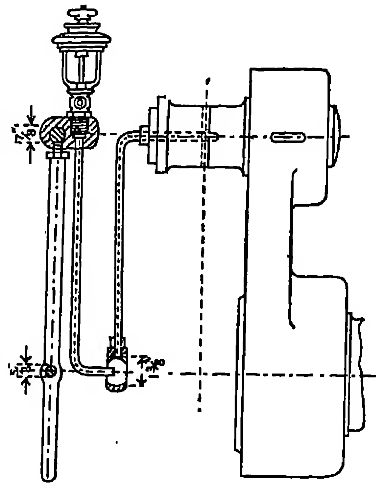


Fig. 5

Adjustment is provided for in bearings where more wear than can be

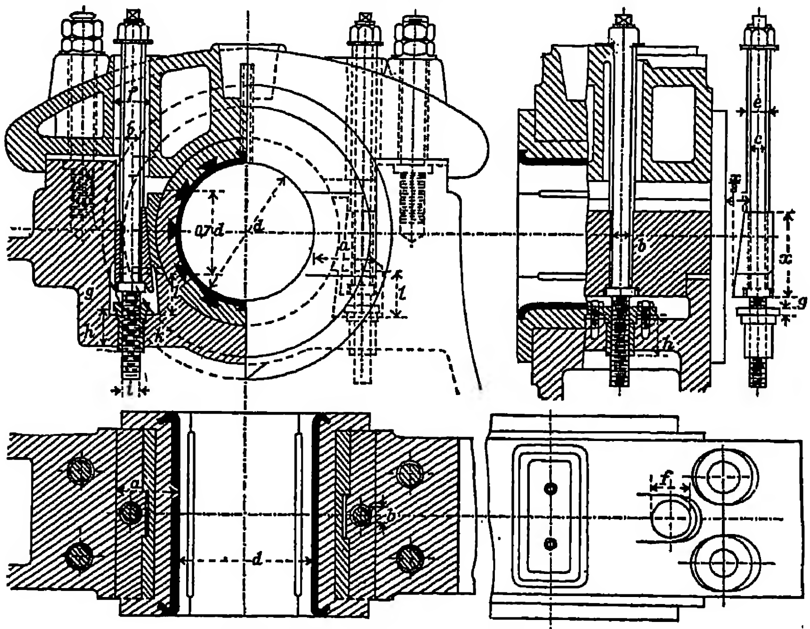


Fig. 6.—Bearing with Brasses in Four Parts

permitted is likely to occur, or where certain accuracies are necessary. In fig. 6 is shown a large crank-shaft bearing with the means of adjustment

of the various parts. In important cases bearings are cooled by water circulation. The spindles of machine tools run with a very thin film of oil, and should have means of close adjustment which do not interfere with the alignment. The wheel spindle of a grinding machine is shown in fig. 7, the adjustment for side play being made by drawing the bearing, which is split longitudinally, and tapered on the outside, into the internally tapered bush, by means of the square-threaded nuts on the end of the bearing bush. With thin films bearings must run distinctly warm, and if the oil supply is efficient will do so without injury.

Thrust Bearings.—The axial thrust on a journal may be taken on flat collars (as in fig. 7) or by any surfaces of revolution provided that the taper is not too slight; but as flat surfaces cause the least frictional loss and are simple they are the most usual form. As the frictional loss increases with the radius of the collar, several collars may be used to carry the load,

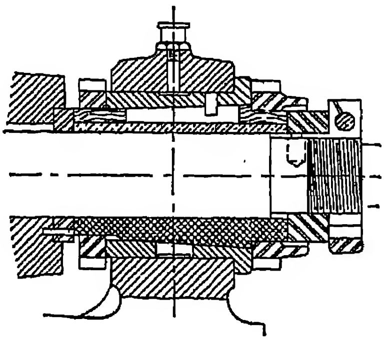


Fig. 7

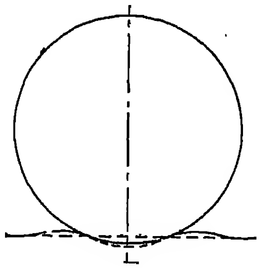


Fig. 9

as in propeller-shaft thrust blocks, which are water-cooled in the larger sizes. The collars are solid on the shaft, and the blocks capable of adjustment so that the total thrust can be distributed over the various surfaces. The use of such thrust blocks is being superseded by that of the Michell thrust (fig. 8), in which the bearing consists of plates arranged to pivot, so that they can take up automatically positions which give a tapered shape to the oil film, enabling it to be easily established and maintained; this reduces the necessary area and friction so that only one collar is employed.

Balls and Roller Bearings.—In cases where high efficiency is required, roller and ball bearings are employed. A ball touches the race at a point considered geometrically, but under the elastic distortion due to the load this (fig. 9) becomes a small area. If the instantaneous axis of the ball relatively to the race lies in the tangent plane at the point of contact, the ball has a pure rolling motion. This involves very little friction, which is due to the slipping of the strained surfaces as the state of distortion moves in the ball and in the race. If, however, the instantaneous axis be inclined to the tangent plane, the motion consists of a pure roll combined with spin about the normal, the latter action introducing more

friction and wear, though both are yet very small. Direct tangential slip of the surfaces (the instantaneous axis not passing through the point of contact) causes rapid wear and cannot be permitted. At high speeds it may arise from gyroscopic action.

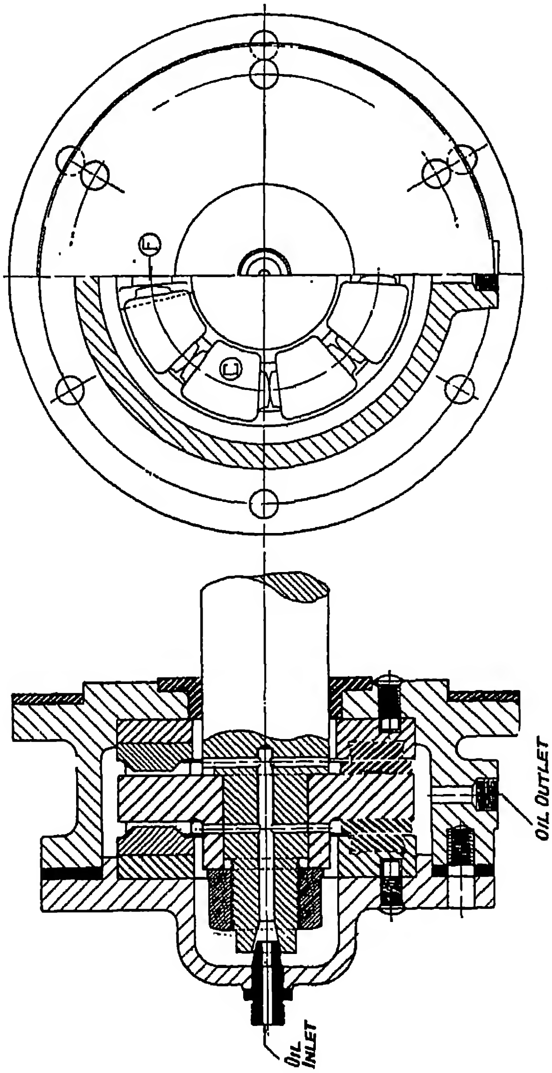


Fig. 8.—Michell Thrust Block

In fig. 10 is shown a spindle carried on two (unlike) ball bearings, the adjustment being made by the nut D, moving the loose cone E until contact takes place. Considering the races AB and C fixed and the spindle revolving, as there will be no slipping at A or B, AB is the instantaneous axis of the ball in space, and hence H and E being fixed points as regards its motion on the cone, HE is its instantaneous axis relatively to the cone. The ball in the case, therefore, when the vertex of cone E is at J—a point

distinct from H—both rolls and spins at the points A, B, and E. If the conical race E had its vertex at H, there would be pure roll only upon the cone. This type of race has three-point contact. The bearing at the other end is a two-point bearing with races of curved axial section, the tangent planes at the ball contact making the axis in M and K, giving spin at the point of contact having least friction.

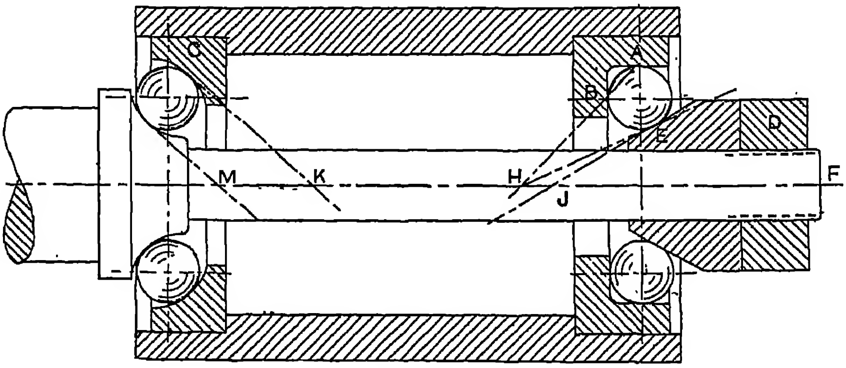


Fig. 10

As the area of contact is very small, it is necessary, whatever the type of construction, that all balls and rollers in a race must be exceedingly true to shape and size (one ten-thousandth of an inch limit on the diameter is commercial), and the races closely true surfaces of revolution, and to carry much load all must be of hardened steel. The difficulties of production have led to the growth of specialist firms, who manufacture bearings of the self-contained type, both journal and thrust. In fig. 11 is shown a section of a double-row journal bearing by Messrs. The Skefko Company, the outer race being in the latter case spherical internally, which permits the shaft angular freedom and has the advantage that a pair of bearings automatically align themselves. In usual practice the balls are separated by softer metal cages, which prevent two hardened steel balls from rubbing one another. No adjustment for wear is provided in these bearings, as under normal conditions practically no wear occurs. Journal bearings of these types can only carry a comparatively small end load, and provision must be made in their use for preventing this arising. Thus one of the pair of bearings only may be locked in position, and end freedom must be permitted to the other, so as to prevent end thrust due to uneven rise of temperatures. In fig. 12 is shown such an arrangement, the outer races for the balls being fixed in the loose pulley, while one inner race is fixed to the shaft, the other being free to slide end-ways. The caps make provision for excluding dust, which is important. The provision for sliding is made

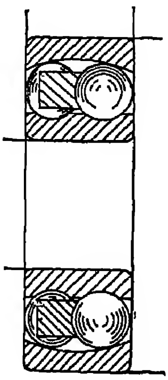


Fig. 11

at the shaft, as, in order to prevent creep, the bearings must be firmly fixed to the part (male or female) which revolves. In fig. 13 is shown a thrust race by Messrs. The Skefko Manufacturing Company.

Rollers touch the surface of their races along a line, which becomes under load a long narrow area, and they are therefore capable of carrying much higher total loads than balls are. Long rollers have a tendency to travel endways in the bearing and cut through their cages, so that cheese-shaped rollers are employed.

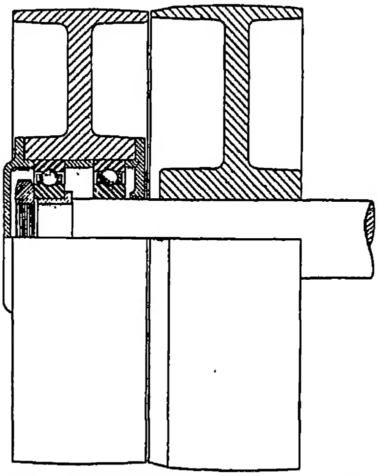


Fig. 12.—Ball Bearing mounted in Loose Pulley

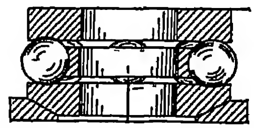


Fig. 13—Skefko Single-thrust Bearing

Roller thrust bearings (e.g. for swing bridges) are ideally made with taper rollers and conical (flat as a particular case) races, all vertices being coincident, but in practice cheese-shaped rollers between parallel plates have been found quite satisfactory.

Slides.—As the movement of a slide reverses, the conditions for lubrication are not so favourable as those in bearings, and the wear is greater. Sometimes a slide can be arranged to run in an oil bath, but usually lubrication is at best by rollers and oil pads. In fig. 14 are shown conical rollers for lubricating the ways of a planing machine. A slide and its ways should preferably be arranged that they should overrun; if the limits of the movement are constantly the same (as with the crosshead of an engine, unless there is overrun), the wear leaves a ridge which may cause trouble when an adjustment (e.g. of the main bearings) is made.

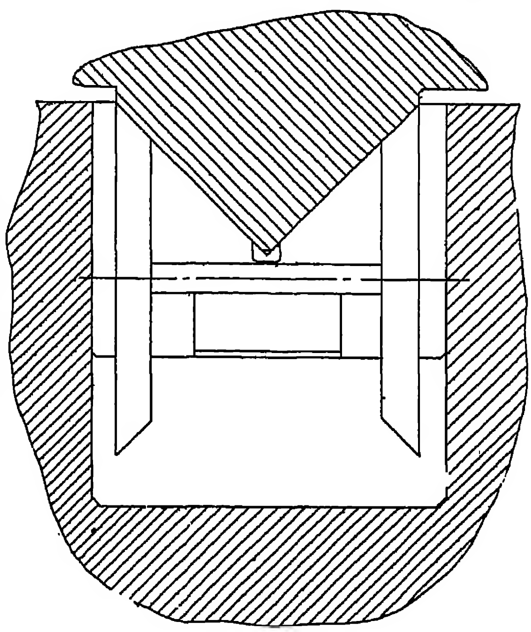


Fig. 14

In planer and some grinding machine ways no adjustment is provided for wear, which—as regards the cross section—tends to make a closer fit. Longitudinal wear tends to make the ways convex, which is corrected in planers by springing the bed. In fig. 15 is shown a machine-tool slide with the means for taking up the wear—by taking out the bolts and strip, scraping the base of the strip, and replacing.

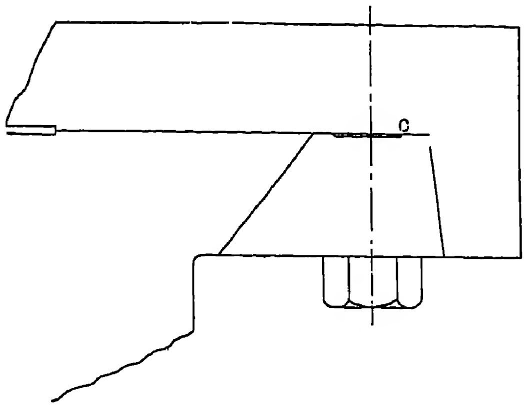


Fig. 15

Ball slides, as shown in fig. 16, are occasionally employed. The balls should be held in a cage, which travels at the rate of the ball centre—which is half the speed of the moving slide when the race is the same in both elements of the slide.

Screw Pairs.—In the specification of a screw both the pitch and the

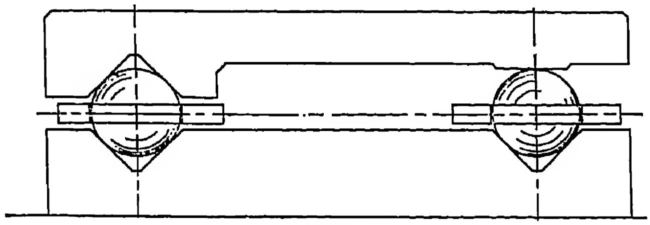


Fig. 16

axial cross section are required, which presents scope for great variety. In a geometrical screw thread or helix the pitch p, the diameter d, and angle of helix α (fig. 17) are connected by the equation

$$p = \pi d \tan\alpha,$$

and if the helix be supposed drawn upon a cylinder, the latter may be supposed developed, the helix becoming a straight line. In fig. 18a is shown a square-threaded screw, in which the section ABCDEF consists of lines parallel to and perpendicular to the axis, AB, BC, and CD being frequently

but not necessarily equal. If the screw and nut be (as usual) of different materials, AB and CD should not be equal but proportioned to secure strength. The screw may be single-threaded right-hand, the pitch being

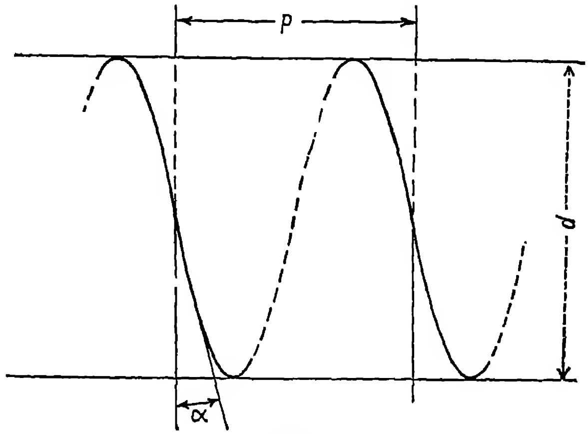


Fig. 17

AE, whereas in fig. 18*b* a triple-square-threaded (or three start) left-hand screw is illustrated. The pitch is now 3 × AE. The pitch is sometimes termed the lead, and the term pitch used for the sectional pitch. Thus the pitch of the flutes of a point drill or spiral reamer is usually termed

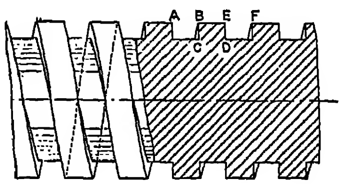


Fig. 18*a*

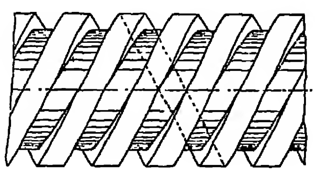


Fig. 18*b*

the lead. In actual screws, such as these, the thread angle varies, being greater at the root than at the outer part of the thread, and may be taken as a where

$$p = \pi(d - h) \tan a,$$

h being the depth of the thread.

Efficiency of Screw Pairs.—If such a screw be employed to raise a load W—as in a screw jack—there will be a normal action δR on an element of the thread, accompanied by a tangential force $\mu\delta$R along the thread, the sum of the resolved vertical components of these being equal to the total load given (fig. 19):

$$W = R(\cos a - \tan\phi \sin a),$$

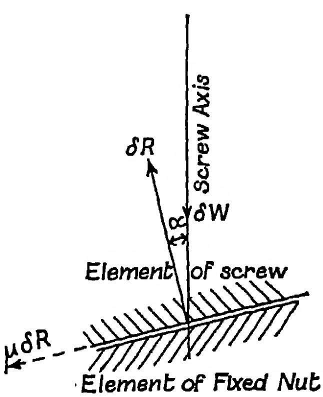


Fig. 19

where α is the mean inclination of the thread and ϕ is the angle of friction, so that $\tan\phi = \mu$. The summation of the moments of δR and $\mu\delta R$ about the axis gives the total torque L, which will just cause the load to rise, and hence

$$L = R(\sin\alpha + \tan\phi \cos\alpha)r.$$

Hence $$\frac{L}{W} = \frac{\sin\alpha + \tan\phi\cos\alpha}{\cos\alpha - \tan\phi\sin\alpha}r = r\tan(\alpha + \phi).$$

Comparing this with the value of L for $\phi = 0$ the efficiency of the screw is given by

$$\eta = \frac{\tan\alpha}{\tan(\alpha + \phi)},$$

which depends mainly upon the angle α.

The maximum value of η is given by $\frac{\delta\eta}{\delta\alpha} = 0$, or $\alpha = \frac{\pi}{4} - \frac{\phi}{2}$, which, in practice, is nearly 45°. The values are illustrated by the table from Sir Alexander Kennedy's *Kinematics of Machinery*.

Tan α.	Efficiency $= \frac{\tan\alpha}{\tan(\alpha+\phi)}$.				
	$\tan\phi = 0{\cdot}01$.	$\tan\phi = 0{\cdot}02$.	$\tan\phi = 0{\cdot}03$.	$\tan\phi = 0{\cdot}10$.	$\tan\phi = 0{\cdot}20$.
0·000	0	0	0	0	0
0·025	0·713	0·555	0·458	0·203	0·112
0·050	0·829	0·706	0·622	0·331	0·196
0·075	0·883	0·741	0·699	0·429	0·270
0·100	0·906	0·828	0·766	0·495	0·325
0·125	0·924	0·858	0·805	0·552	0·376
0·150	0·935	0·877	0·829	0·592	0·414
0·175	0·943	0·893	0·849	0·627	0·450
0·200	0·950	0·904	0·865	0·656	0·480
0·225	0·954	0·913	0·876	0·678	0·505
0·250	0·958	0·920	0·886	0·698	0·527

If the axial force move the screw through the block, the direction of the frictional force will be reversed, and the efficiency becomes $\frac{\tan(\alpha - \phi)}{\tan\alpha}$, and for self locking α must be less than ϕ.

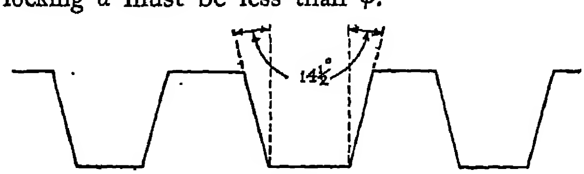


Fig. 20

Screw Threads.—For many purposes the square-threaded screw is inadmissible or unadvisable, and the side of the thread is inclined. As

sharp corners are liable to damage, and are a source of weakness, they are removed in most threads. Fig. 20 gives the " acme " thread shape, and fig. 21 a ratchet type, employed when the load is in one direction.

When the thread section is inclined at an angle β the elementary

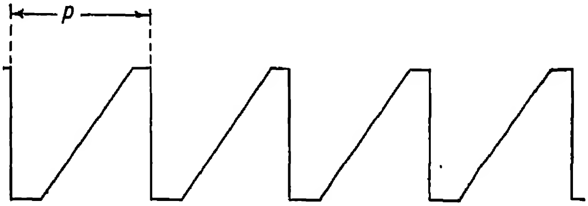


Fig. 21

axial force δW leads to an elementary normal force $\sec\beta . \delta W$, and hence the efficiency is given by the same expression as for a square thread, if $\mu \sec\beta$ be substituted for μ.

In fig. 22 is given the Whitworth screw-thread section, the angle between

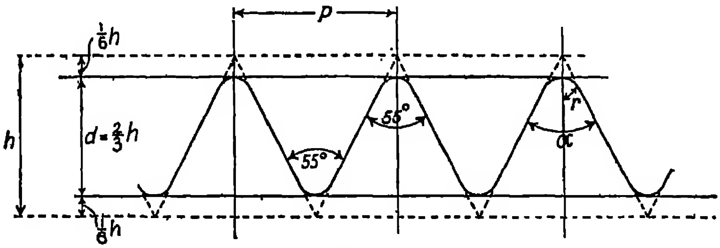


Fig. 22

the sides being 55°, and $\frac{1}{6}$ of the depth rounded off from the top and bottom corners. The Swiss (Thury) section is similar, the inclination of the sides being $47\frac{1}{2}°$ and the rounding $\frac{1}{6}$ of depth, the British Association thread being the same but with $\frac{2}{11}$ rounding. The Sellars' or American thread section is given in fig. 23.

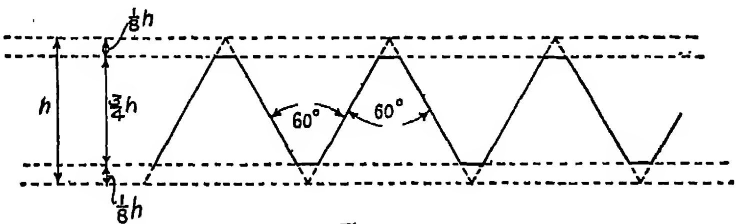


Fig. 23

The fit of screw threads should be upon the straight portion of the section, and clearance between nut and screw given elsewhere. Thus a tap should measure oversize on the tops of the threads. The British Engineering Standards Association have arranged tolerances upon the B.S.F. screw threads.

Differential Screw.—When a fine movement is necessary, and the screw sustains a heavy thrust, a differential screw (fig. 24) may be employed. Here the part *b* receives, for one turn of the screw *a*, a movement which is

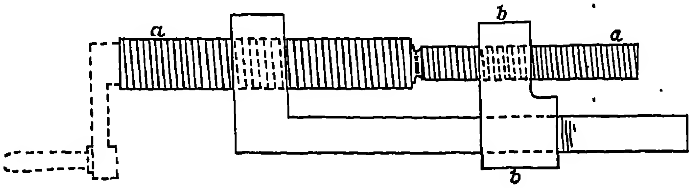


Fig. 24

the difference of the pitches of the two screw threads, which difference may be made small, although both of the screw threads are made large enough to carry the load.

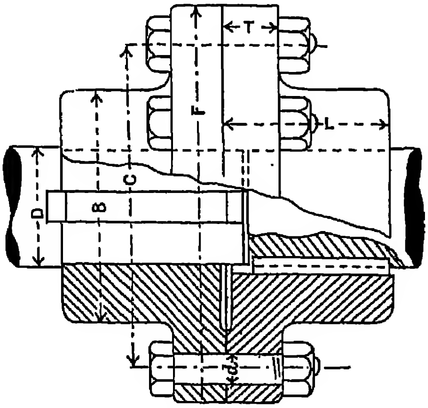


Fig. 25

Couplings.—The simplest case in which one shaft is driven from another is when they are co-axial, the connection being then termed a coupling if intended to be permanent, and a clutch if the connection is to be made and broken frequently. Flanged, split-muff, and Sellars' couplings are shown in figs. 25, 26, and 27 respectively. In the first the flanges are keyed to the shafts and connected by bolts, the shear in which transmits the torque; the flanges should be centred in one another as shown, and the bolts a turned fit. Keys or overlapping shafts are usually unnecessary with split muff or Sellars' couplings, in the latter of which the grip on the shafts is obtained

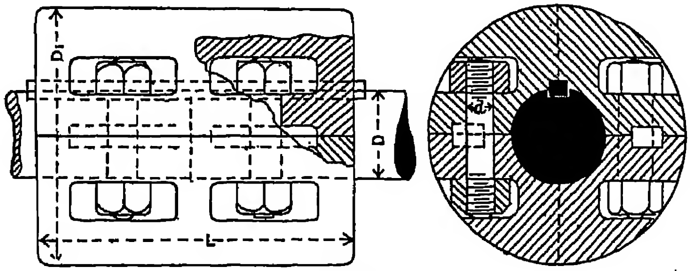


Fig. 26

by drawing the split taper bushing into the hollow sleeve by means of the longitudinal bolts.

Bored parts are fixed to shafts by keys of various types, castellations, or taper pins. Types of key fitting are shown in fig. 28, saddle key;

fig. 29, flat key; fig. 30, sunk key with round ends; fig. 31, Woodruff or semicircular keys. If the bored part is to slide along the shaft

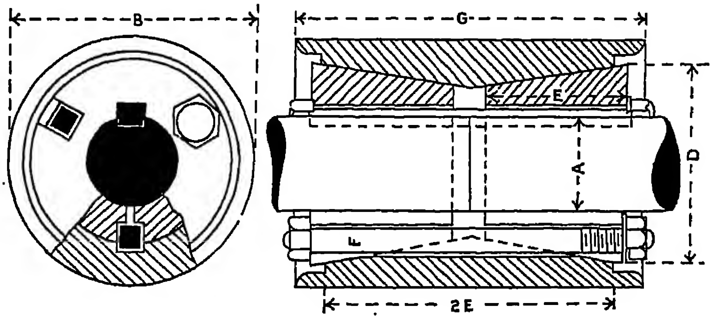


Fig. 27

without turning, the key, then termed a feather, is fixed to the sliding part and moves along an elongated keyway. Woodruff keys are used

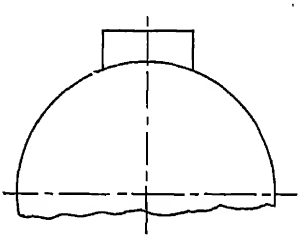


Fig. 28

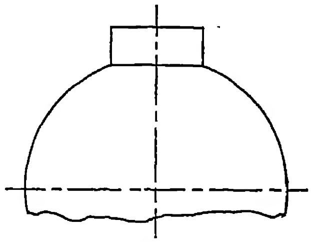


Fig. 29

extensively in machine-tool construction. These keyway ways are simple to produce and the fit easily made, but the shaft is very considerably weakened. If a long key is needed a sequence of Woodruff keys are fitted. The shaft strength is reduced least by the castellated fit. Usually the sides AB and CD of the projections (fig. 32) on the shaft are made parallel. If the parts are not subjected to the distortion involved in hardening, the diametral fit is upon the outer parts BC, and obtained from the drift shape and by circular grinding the shaft, but for hardened parts the diametral fit should be upon DE, and obtained by grinding the inner diameter of the hole and the bottom of the castellation in the shaft.

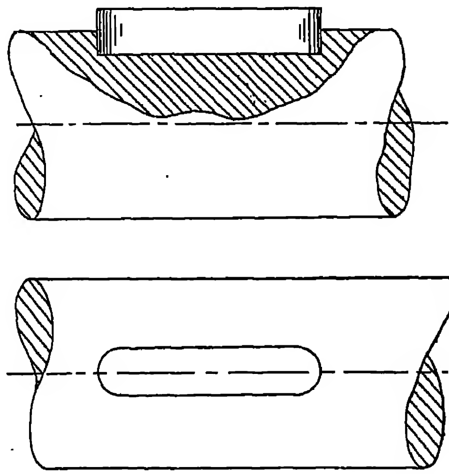


Fig. 30

The connection between parts and shafts may also be made by taper pins (fig. 33), the taper being standardized at $\frac{1}{4}$ in. per foot, at which the elasticity of the material is sufficient to retain the pin under considerable vibration.

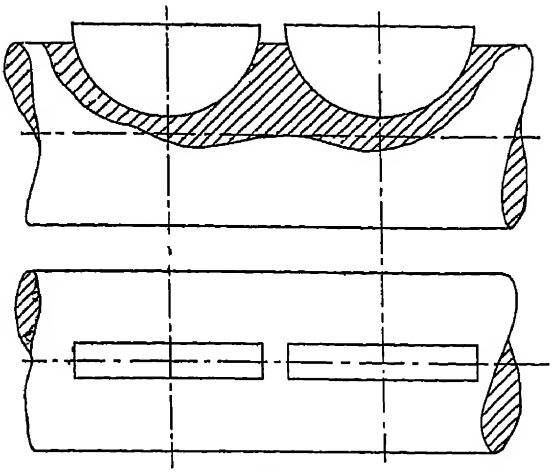


Fig. 31

Where axiallity of the parts is of great importance the parts should fit upon a taper, and if this be of a certain fineness the hold will be elastic. For drills which need to be retained in the socket elastically against small axial forces (such as their weight) tending to extract them, the Morse system of tapers is employed, while for end-milling cutters a finer taper, standardized by Messrs. Brown & Sharpe, is used.

Cotters and Gibs.—Parts to be forced into contact may be connected by cotters, gibs being used if necessary. In fig. 34 the cotter, when driven into its slot, forces the parts upon which it bears in the direction enforcing the contacts desired. To check the tendency of the lower outer part to bend outwards under the friction as the cotter is driven

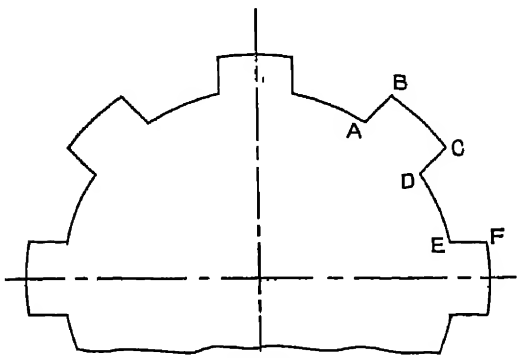


Fig. 32

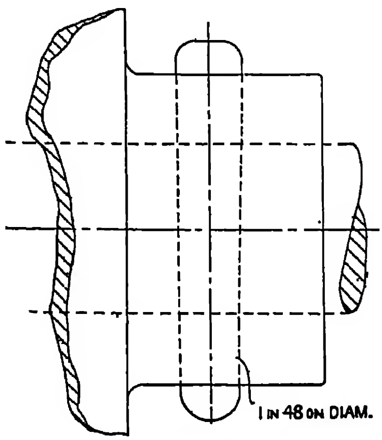


Fig. 33

home, a gib may be employed as in fig. 35. The cotter may be of rectangular section, or the ends may be rounded. Taper pins may be used as cotters, in which case the holes reamed for the taper pin should have clearance given to them to produce the cotter action.

Oldham Coupling.—Shafts whose axes are parallel but slightly out

of line can be connected, so as to rotate in unison, by means of an Oldham coupling (fig. 36), which consists of flanges with a keyway cut across each, keyed to the shafts, and an intermediate piece carrying a key on each side, the keys being at right angles. By the sliding freedom provided, the intermediate piece will fit both flange faces, provided the keyways in the shaft flanges are perpendicular, and will therefore drive the second shaft in unison with the first.

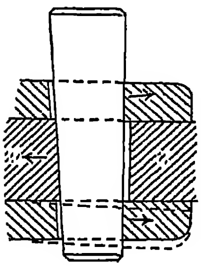


Fig. 34

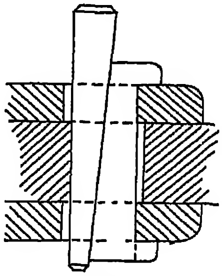


Fig. 35

Clutches.—In a claw coupling, the force is transmitted by the face of the jaws, one part sliding into the other for connection. If there is much initial load, which may be due to inertia, such a clutch cannot be engaged when running, nor can it be disengaged, owing to the friction between the jaws arising from the driving force. If the jaws be cut on an angle, so that their faces are screw threads, they can be disengaged if the angle is nearly equal to the angle of friction, and a lock is then necessary to keep them securely engaged. If such a coupling is to be engaged under a light load, ample clearance is desirable.

A friction clutch is capable of engagement under load, taking the load up gradually, the cones (in a cone clutch) slipping until sufficient axial force

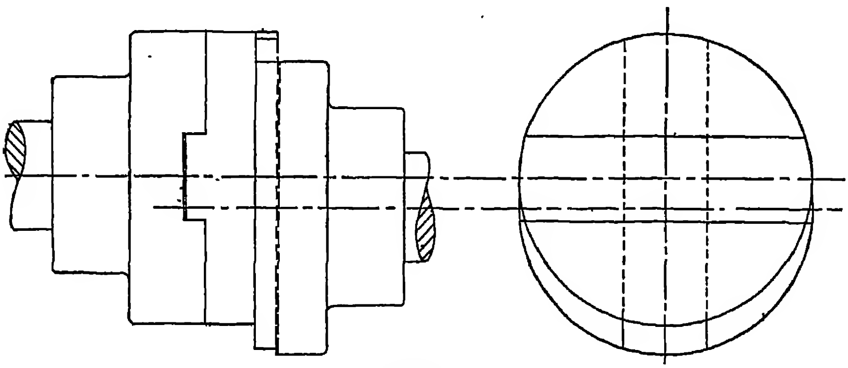


Fig. 36

is applied. It is essential that clutches should disengage with certainty, so that the angle of a conical clutch must not be too small; 8° to 14° is the customary range. To increase the friction, one or both cones may be lined with leather or other material.

A plate clutch consists of a series of plates (or a single plate) keyed to one shaft, alternating with plates keyed into a cylinder attached to the second shaft. The plates are free to move axially through a short distance, so that an axial force produces a frictional torque at each plate

surface. Such a clutch is easily lubricated, and the surfaces of the plates accordingly wear well and engage smoothly.

Belts.—Motion can be transmitted between shafts by means of belts, ropes, or chains, which may be considered as flexible gearing; by friction or toothed gearing; by cams; or by linkage connections. Of these, belts, ropes, and friction gearing may involve slip, but have the corresponding advantage of yielding under overload.

Leather belting varies greatly, dependent on the quality of the hide as a whole and the part of it used. The strips of hide run about 4 ft. long, and are built up by cemented scarfed joints into a continuous piece, and by a similar joint the belt may be made endless, which is desirable in main drives, grinding machines, &c., as it avoids irregularities which are introduced by fasteners at high speeds, and the belt is not weakened by holes for the fasteners. To run well, belting should lie straight when unrolled. Belting is also woven from various materials, or made of impregnated canvas, and such belts run well on plain drives and are desirable in certain atmospheres. For working on loose pulleys or speed cones, leather is to be preferred.

A belt pulley should be at least $\frac{1}{2}$ in. wider than the belt, and may be flat faced or " crowned ", which tends to keep the belt on the largest diameter of the crown. The sides only of the face should be crowned if the belt be wide. Pulleys to be fixed to a main shaft for secondary drives are usually split for convenience, and made of pressed mild steel; composite with sheet-steel rims, round steel shafts and cast-iron hubs; cast iron; or wood. The first class are the most satisfactory in small sizes, the depreciation being low and there being no trouble from the heat generated by a slipping belt.

Secondary machines are usually controlled by friction clutches or by fast-and-loose pulleys. In the former case either the clutch pulley or the main shaft pulley should be crowned, in the latter the fast-and-loose pulleys should be crowned and the main shaft pulley flat. The loose pulley should be a little less in diameter than the fast pulley (so as to relieve the tension in the belt when not driving), with a conical incline to the fast pulley; it should be bushed so as to contain an oil reservoir, and fitted with felt pad or other device for controlling the delivery of the lubricant.

Since the speed of the belt is the same at all points, the rotational speed of the connected shafts is inversely proportional to the diameters of the pulleys upon them, or to a closer degree of accuracy, allowing for the belt thickness t,

$$\frac{n_1}{n_2} = \frac{\omega_1}{\omega_2} = \frac{d_2 + t}{d_1 + t},$$

where n_1 and n_2 are the revolutions of the shaft in any time, ω_1 and ω_2 their angular velocities, d_1 and d_2 the pulley diameters. As the belt is stretched more on the tight side than on the other, the speed of the driven shaft is rather less than the above equation gives, owing to the slip this necessarily

introduces. If the belt be crossed so that the shafts rotate in opposite directions, the angular velocities may be considered to be of opposite signs.

Cone Pulleys.—Cone pulleys are the simplest method of speed variation in regular use. The belt must fit upon each pair of the steps of the cone, otherwise the belt soon receives such permanent stretch that it will not drive upon the pairs of steps which require the shortest belt. If the belt be crossed the condition is that the sum of the diameters of each pair of steps is the same. With open belt the methods of determining the pulleys to give selected speeds are indirect with the exception of the following (which is due to the writer), where, if x and kx be the pulley diameters required, k being one of the required speed ratios

$$x = \frac{1}{k-1}\frac{P}{Q}\left[1 - \frac{P}{Q^2 b}\right],$$

where P is the difference between the belt length and twice the distance between the shaft centre lines, and is constant for all the steps and is selected to give a suitable size to one of the steps, and $Q = \pi\frac{k+1}{k-1}$ is different for each speed ratio.

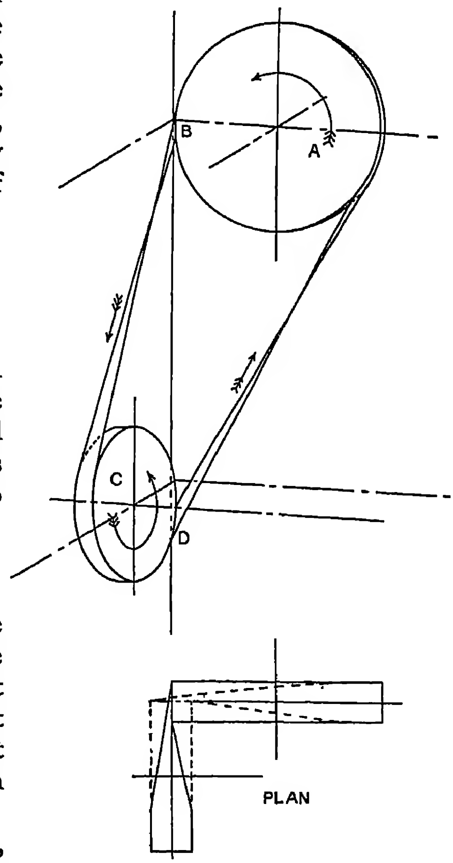


Fig. 37

Skew Belt Drives.—If the shafts to be connected by belt are not parallel, the sole condition that the belt should run correctly is that it should leave each pulley at a point in the plane of the pulley to which it runs.

This is illustrated in fig. 37, which shows a half cross drive between a pair of non-intersecting shafts at right angles. The pulleys are arranged so that the belt leaves the pulley A at the point B, which is in the central plane of the pulley C, and it leaves the pulley C at the point D, which is in the central plane of the pulley A. In the lay out, upon the determination of the pulley sizes, they are set as in the plan view, and the belt then arranged as shown.

If the direction of rotation of the driving shaft be reversed (e.g. as happens occasionally with gas-engine drives) the belt will fall off the pulleys This can be avoided by the introduction of idler pulleys into the system, as illustrated in fig. 38, where two idlers are employed, and the belt will

run in either direction. An endless belt can be used, one of the idlers being made vertically adjustable to take up the stretch. A single idler pulley, placed askew, can be used, but the two parts of the belt run very closely together at the idler, and in adjustment the idler pulley must be twisted as well as moved bodily. In all twisted belt drives the outer edges of the belt are slightly more stretched than the centre line: that both sides come in contact with pulleys has little effect.

Power transmitted by a Belt.—The friction between a belt and the pulley acts over the whole arc of contact, ABCD in fig. 39, of the belt, and for the maximum difference between the tension T_2 in the tight side and T_1 in the slack side of the belt, the coefficient of friction must everywhere have its maximum value μ. Considering an element BC ($a\delta\theta$) of the belt, a being the radius of the pulley and θ the angle AOB, where O is the pulley axis, its mass is $ma\delta\theta$, m being the mass per unit length of the belt, its acceleration is $\frac{v^2}{a}$, where v is the velocity of the belt, and it is acted upon by

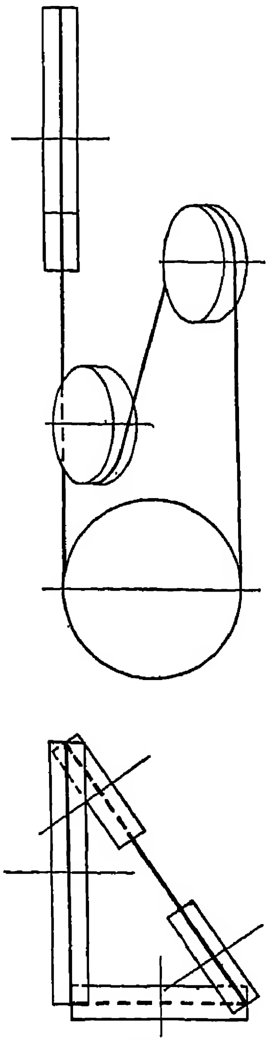


Fig. 38

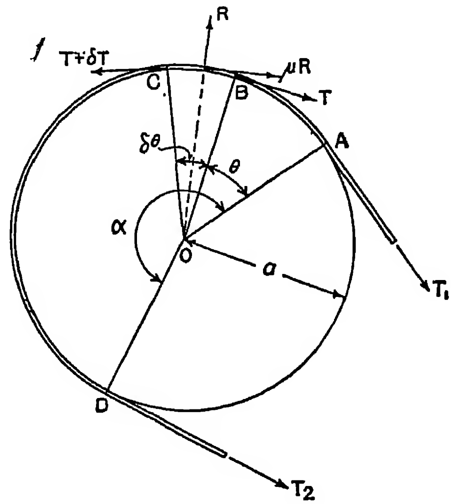


Fig. 39

the tensions T at B and T + δT at C, by the normal reaction R and the consequent friction μR, all in the directions indicated. Resolving along the radius and tangent gives, putting the acceleration into gravitational units,

$$\frac{ma}{g}\delta\theta\,\frac{v^2}{a} + R - T\delta\theta = 0,$$

and
$$\delta T - \mu R = 0.$$

The elimination of R from which gives

$$\frac{mv^2}{g}\delta\theta + \frac{\delta T}{\mu} - T\delta\theta = 0,$$

whence $$\left(T - \frac{mv^2}{g}\right)\mu d\theta = dT,$$

or $$T - \frac{mv^2}{g} = Ae^{\mu\theta}.$$

The values of T_2 and T_1, being obtained by putting $\theta = \alpha$, the angle ABCD of embrace of the pulley by the belt, and $\theta = 0$, the elimination of the constant A gives

$$\frac{T_2 - \frac{mv^2}{g}}{T_1 - \frac{mv^2}{g}} = e^{\mu\alpha}.$$

As the term $\frac{mv^2}{g}$ thus expresses the effect of the belt velocity on the tension it is sometimes termed the centrifugal tension. For slow speeds its effect may be omitted, the relationship becoming

$$T_2 = T_1e^{\mu\alpha}.$$

The horse-power transmitted is $(T_2 - T_1)v$, or $T_2(1 - e^{-\mu\alpha})v$ for slow speeds, and

$$\left(T_2 - \frac{mv^2}{g}\right)(1 - e^{\mu\alpha})v$$

when the speed is high. The speed at which the maximum horse-power is transmitted is therefore given by

$$T_2 = 3\frac{mv^2}{g}.$$

The greater tension T_2 is controlled by the economical life of the belt, which is found to limit it to about 80 lb. per inch width for single belting, although such belting has about 1000 lb. breaking load.

The density of belting is about 0·11 lb. per inch width per foot length, so that $\frac{m}{g} = 0{\cdot}0035$ per inch width of belt. Running the belt at 80 lb. per inch width maximum tension gives, for the velocity at which the maximum horse-power is transmitted,

$$v = \sqrt{\frac{80}{3 \times 0{\cdot}0035}} = \sqrt{7620} = 87{\cdot}3 \text{ ft. per second},$$

or 5250 ft. per minute.

For high-speed power transmission, therefore, a belt whose working tension is high compared to its weight should be employed. Belts of thin

sheet steel—in which the effect of the stress caused by the bending must be considered—have proved successful in such transmissions.

Rope Drives.—When ropes, cotton or hemp, are used the pulleys are grooved to keep them in running position, but the principles of application are the same as in the case of belts. For large drives, in which a pulley has a number of rope grooves, a rope may be employed for each groove, or a single continuous rope used, running several times round the system of driving and driven pulleys and fitted with a single-tension pulley, weighted to maintain a correct tension (in a slack part of the rope) and to take up stretch.

Chains.—The development of chain transmission arose from the cycle in which link chains fitted with rollers are usually employed. The chain wheels are actually polygons, the sides being links and the corners the pin joints of the chain on the wheel, and hence the transmission does not give strictly uniform motion. The stretch of chains in use has led to the invention of the silent chain. The general velocity ratio transmitted by chains is positive, being that of the number of teeth in the sprocket wheels, and where the small irregularities previously mentioned do not matter, they form a convenient means of positive drive over a distance.

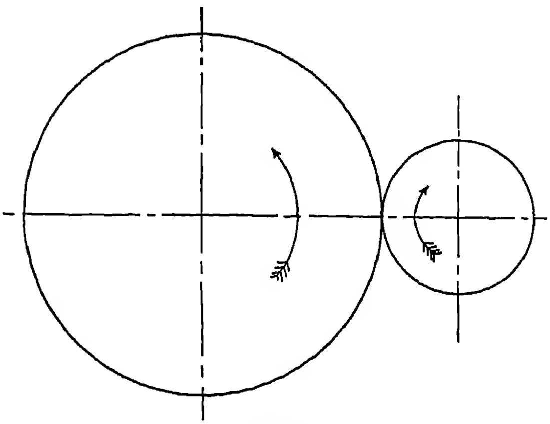


Fig. 40

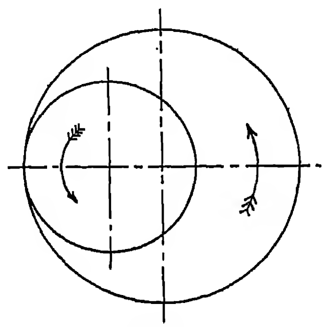


Fig. 41

Chain of the ordinary type is used in the Weston block. The upper pulley has a double set of chainways in it, containing m and n link recesses respectively. The hoisting chain runs round the first part of pulley, then round the lower pulley, and back in the reverse way round the second part, and finally rejoins itself. If a length of m links be drawn off the pulley, raising the lower pulley, n are simultaneously wound on, so that the lower pulley rises $\frac{m-n}{2}$ link-lengths, giving a velocity ratio of $\frac{2m}{m-n}$. The shape of the link pockets is important, as the block must be safe against slip. The efficiency is low (about 25 per cent), but the high velocity ratio, due to the differential action and its simplicity, renders the device valuable.

Friction Gearing.—If a pair of shafts intersect, either at infinity or at a finite point, they can be connected by surfaces which roll upon one another without slipping, and so can transmit power by friction at the contact. The generators of these friction surfaces pass through the point of intersection of the shaft axes, and the contact is along a generator. If the axes do not intersect or the surfaces are otherwise formed, slipping takes place at the surfaces, causing wear. The force transmitted is limited by the product of the coefficient of friction and the force with which the surfaces are pressed together. To secure a high frictional force, one (or both) of the surfaces is faced with leather or other material having a high coefficient of friction, and, as slip will take place under overload, the driver should be softer than the driven gear, so that the wear is distributed over its surface and not confined to one place. Friction gearing, consisting of two cylinders with external contact, is shown in fig. 40, and with internal contact in fig. 41, the length of the cylinders not being indicated. In fig. 42 is shown a pair of conical friction gears. These all transmit

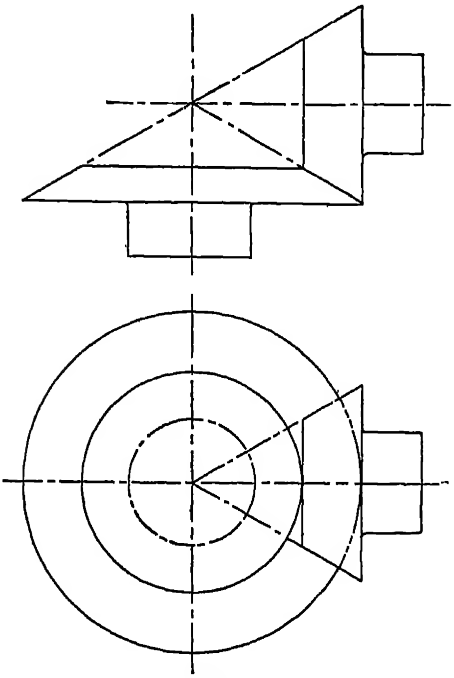


Fig. 42

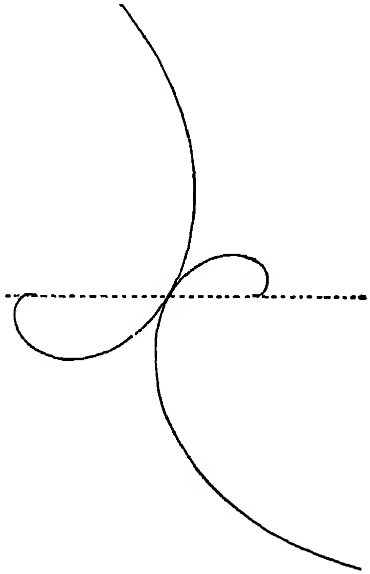


Fig. 43

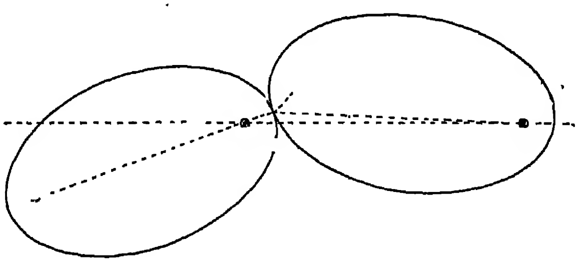


Fig. 44

velocity uniformly from one shaft to the other. Other surfaces may be employed, giving a changing velocity ratio between the shafts. Illustrations of this are given in fig. 43, showing a pair of logarithmic spirals of equal angle pivoted at their origins, and in fig. 44, which shows a pair of ellipses pivoted at their foci.

A friction gear which serves as a convenient device for speed variation is shown in fig. 45. Here the leather-faced disc A, arranged to slide with a feather along the shaft BC, drives the disc DE, which is held up into contact with A by the spring F. If BC be driven at a constant speed and the disc A set out at the edge D of the driven disc, the latter receives its slowest speed. As A is moved towards the centre of disc DE the speed of the latter increases, and when A passes over to the side E of the driven disc, the direction of rotation of the latter is reversed. Here the friction

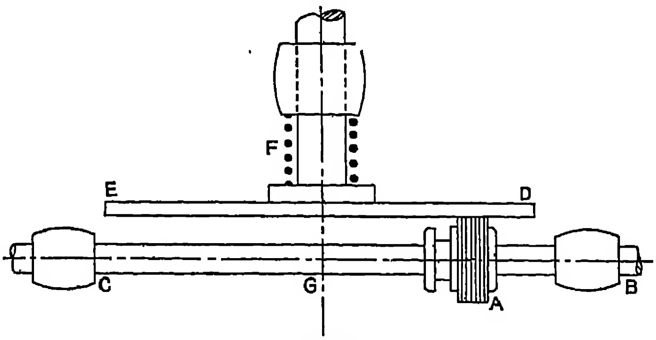


Fig. 45

surfaces are not generated by lines passing through G, the point of intersection of the shafts, and hence slip occurs at the surfaces in contact, causing wear.

If the axes of the shafts do not intersect in a point, motion may be uniformly transmitted from one to the other by the line contact of a pair of hyperboloids (of one sheet) of revolution, but in addition to the rolling contact there is tangential slip of the surfaces along the common generator.

Toothed Gearing.—If a positive drive is requisite, the friction surfaces must be replaced by toothed surfaces giving the same motion. The imaginary surfaces, which correspond to the friction surfaces, are termed the pitch surfaces of the gears, and upon them the teeth are formed. In practice toothed gearing is almost invariably concerned with the transmission of uniform velocity between the shafts, the teeth in a gear being all of one shape, spaced evenly. The uniformity of transmission is secured by the evenness of the spacing of the teeth, and by the shape of their surfaces.

Circumferential Pitch.—Spur gearing constitutes the simplest form, the pitch surfaces being right circular cylinders. The action of gearing of this nature is completely represented by the action in any plane perpendicular to the axis, and hence the pitch surface is termed the pitch circle. The pitch of the teeth is the distance between the points where similar

tooth outlines cut the pitch circle, as measured along the pitch circle. Thus if a gear have m_1 teeth of circumferential pitch p, the circumference of the pitch circle will be $m_1 p$, and hence its diameter d_1 will be $\frac{m_1 p}{\pi}$.

If a second gear B (fig. 46) having m_2 teeth be in mesh with the gear A of m_1 teeth, the pitch must be the same, so that if d_1 be the diameter of A and d_2 of B,

$$\frac{\pi d_1}{m_1} = p = \frac{\pi d_2}{m_2}.$$

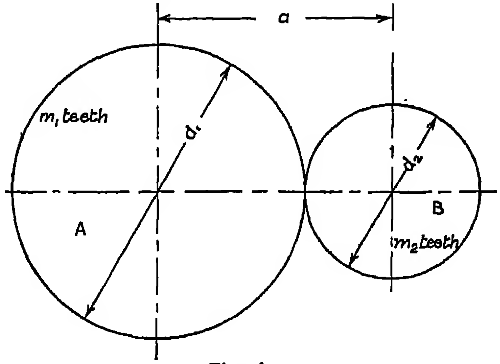


Fig. 46

The distance a between the axes of the gears will be $\frac{d_1 + d_2}{2}$ or $\frac{m_1 + m_2}{2\pi}p$. If the pitch be given in inches or in simple fractions of an inch, since m_1 and m_2 must be the whole numbers, the centre distance a will be incommensurable and difficult in use, both in design and production.

Diametral Pitch.—To avoid this difficulty the pitch—then termed diametral pitch Π—may be expressed as the number of teeth per inch of diameter of the pitch circle. Thus, in the case of the pair of gears considered,

$$\frac{m_1}{d_1} = \Pi = \frac{m_2}{d_2}$$

and $$a = \frac{1}{2}(d_1 + d_2) = \frac{1}{2\Pi}(m_1 + m_2),$$

so that Π being simple arithmetically, d_1, d_2, and a are simple arithmetically.

Module.—The dimension of "diametral pitch", being (length)$^{-1}$ is undesirable, and its inverse, a length, is also employed and termed the module. It is chiefly used in metric work, being then the pitch diameter divided by the number of teeth in the gear. As an example, suppose that a gear 12½-in. pitch diameter have 100 teeth, the circumferential pitch $= \frac{\pi \times 12{\cdot}5}{100} = 0{\cdot}392$ in., the diametral pitch is $\frac{100}{12{\cdot}5} = 8$ and the module $\frac{1}{8}$ in.

Addendum.—It is usual for the teeth to project beyond the pitch circle and the spaces between the teeth to extend within it. The circle through the points of the teeth is termed the addendum circle, although its diameter is usually termed the overall diameter, and the circle through the bottom of the spaces is termed the root circle, the difference of radii of the addendum circle and the pitch circle being the addendum, and of the pitch circle and the root circle the dedendum. The part of the tooth shape which is outside the pitch circle is termed the face, and that inside the pitch circle the flank of the tooth. In an involute gear, considered by itself, there is no pitch

circle, and consequently these terms are not definite, but there is an intended pitch circle to which the terms refer. As one gear drives the other, only one side of each tooth shape is actually working, and clearance is arranged between the other sides, and also at the points and roots.

Tooth Action.—In fig. 47 is shown the action between the teeth of a pair of gear wheels of which A is the driver. A tooth of the driver coming into action encounters the tooth D of the driven, contact being at E between the flank of the driving and the face of the driven tooth. As the motion progresses, the point of contact moves outwards along the driving and inwards along the driven tooth, arriving in turn at the points F, G, where the adjacent teeth are shown in contact; the locus of this point of contact through EFG is termed the path of contact. At the pitch point F—where the pitch lines intersect the line joining the "centres"—the contact is on the line of separation of face and flank of the teeth in contact. The arc along the path of contact to the tooth faces at the commencement of the action is termed the arc of approach, and the corresponding arc to the termination of contact the arc of recess.

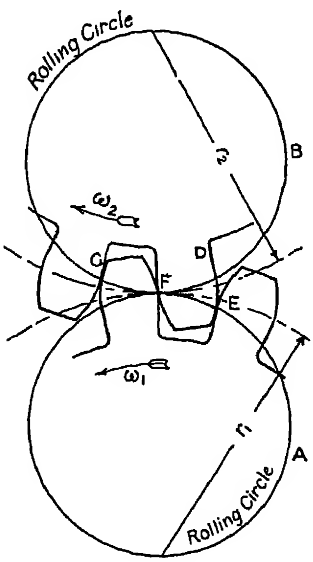


Fig. 47

The tooth action will be identical whether the gears rotate uniformly about their axes, or whether one gear with its pitch line be supposed fixed and the other gear to be fixed to and carried by its pitch circle, as this circle rolls without slipping upon the fixed pitch circle. Since contact is to be maintained, and since the rolling gear is at any moment turning about the pitch point (as its instantaneous axis), the tooth surfaces at a point of contact must be perpendicular to the line joining the point to the pitch point. As the pitch point divides the central distance into parts r_1 and r_2, which are inversely proportional to the angular velocities ω_1 and ω_2 of the driving and driven gears (since the speed of the pitch line is the same considered as an aspect of either gear), the condition for constancy of velocity ratio may be stated to be that the common normal to the tooth surfaces at contact always intersects the line joining the axes at the same point. Subject to this condition, teeth of any shape will work geometrically correctly.

Suppose that one of the gears, let it be B, has its teeth arbitrarily made, of any shape, which may even vary from tooth to tooth, and that this gear be fixed to its pitch circle and this pitch circle rolled upon the fixed pitch circle of A. If a plastic blank be now placed in the position to be occupied by the gear A, the motion of B will form it so that contact will be maintained continuously, the resulting outline of A being the envelope of the

moving body B. Contact in any relative position of the bodies will occur where B touches its envelope, that is, there will be one point of contact for each position of a tooth. In order that contact be maintained through continuous revolutions, it is necessary that the pitch diameter of A should be a multiple of that of B, or else that the teeth are similar in sets which match, or ultimately that the teeth are all similar and divided equally, which is the usual case in practice. Teeth are usually symmetrical, so as to work equally well in either direction, but this is not essential, and where gearing transmits load in one direction only the non-acting faces may be made more sloping with advantage. Thus geometrically the teeth on one gear are the envelopes of those on the other produced in the prescribed relative motion. If the envelope be drawn it may, however, contain loops, which are impractical, and, in the "generating" process described above, the loop would be removed, and part of the surface desired for continuity of contact removed. This is termed "interference", and is to be avoided by a suitable choice of shape for the teeth of B—the "generating" gear. The tooth shape is also to be selected, so that neither it nor the tooth shape generated by it is of a weak shape. Furthermore, any tooth must come into action before its predecessor has finished its contact.

Gear-tooth Systems.—A further limitation arises from the matter of gear manufacture, as the system adopted must be such that gears produced on it will run correctly together whatever the numbers of their teeth, provided only that the pitch is the same. Formerly "cycloidal" teeth, the shapes being composed of a union of epicycloids and hypocycloids, were preferred, chiefly owing to a slightly higher (theoretical) efficiency, but the development of the generating processes of gear manufacture, using the involute as the basis, has transferred the preference to the latter.

Cycloidal Teeth.—If ABC (fig. 48) be the pitch circle, and the circle BDE rolled (without slip) on the outside of it, a point D traces out the epicycloidal curve FDGH, F being the point where D rolls on to the pitch circle ABC; and if a circle BKL be rolled upon the inside of the circle a point K of it will describe the hypocycloid FKMN, the tracing point K being here placed at F when starting, so that the curves join at F, crossing the pitch circle at right angles.

At any instant the tracing point on the rolling circle is moving perpendicularly to the line joining it to the pitch point, which is the point of contact of the rolling circle with the pitch circle; thus D is moving perpendicularly to DB, and when the circle BDE has rolled to the position PQR, D coming to Q is moving perpendicularly to PQ. Now if the circle BDE be also rolled inside another pitch circle RBS, shown in broken lines in fig. 48 and taken as that of a spur gear, the curve traced out by D, a hypocycloid, will touch the epicycloid at D, being also at that point perpendicular to BD. Also in any other position of the rolling circle BDE, as at PQR, the position of the circle RBS simultaneously rolled upon ABC will be TPV, and the curves will touch at Q. Hence teeth of such forms satisfy the conditions for transmitting a constant velocity ratio, and will

work correctly. The path of contact will be made up of two arcs of the rolling circle. In spur gears, therefore, the teeth may be formed by con-

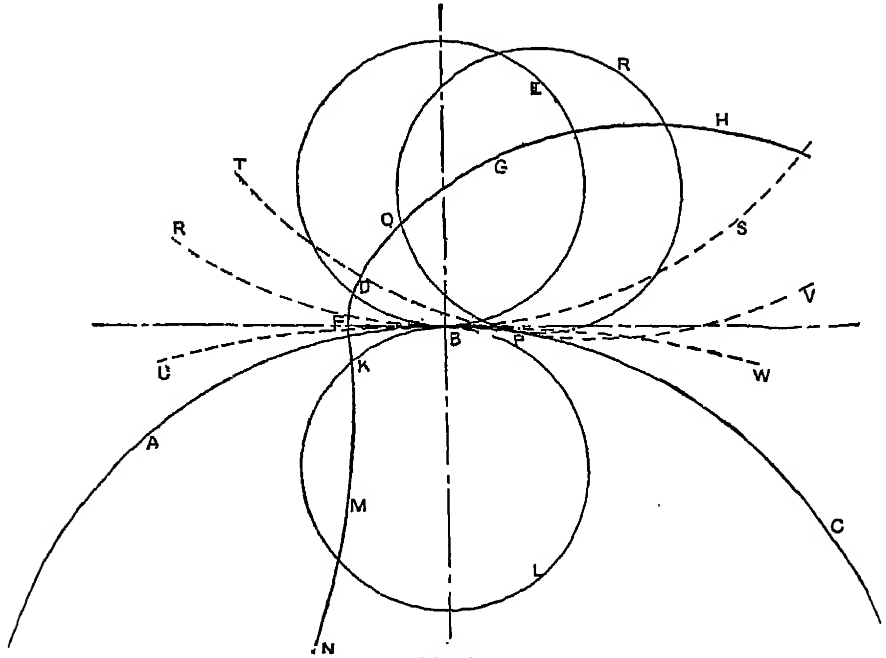


Fig. 48

necting hypocycloidal flanks with epicycloidal faces. If the second gear (fig. 49) be internal, corresponding to the friction gearing of fig. 41, the pitch circle will be of the type UBW, and the teeth will have hypocycloidal faces and epicycloidal flanks. When the pitch circle becomes a straight line (the tangent at B), giving a rack, the tooth shape will be produced by the union of curves, termed cycloids, traced by a point D on the rolling circle BDE as it rolls along a straight line.

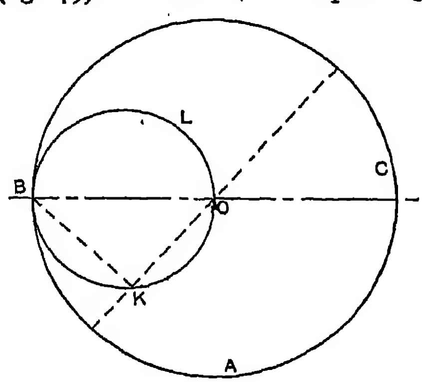


Fig. 49

The tooth shape will depend upon the size of the pitch circle and upon that of the rolling circle, but for a system of gearing one rolling circle must be employed throughout.

If the circle BKL be half the diameter of the circle ABC (fig. 49)—so that the centre O of ABC lies on the circle BKL—the hypocycloid traced out by K will be the diameter OK of the pitch circle, since the angle BKO is a right angle and the motion of K therefore always directed towards O.

The flanks of the teeth will then be radial, so that they are somewhat undercut and therefore weak. This is regarded as the limiting size of the rolling circle with regard to the pitch circle.

The pitch of the teeth being given, the width of the tooth at the pitch line must be less than half the circumferential pitch, owing to clearance being necessary. By drawing the selected epicycloids for the two sides of a tooth, the maximum value of the addendum is obtained by their intersection. This, however, may be lessened, the minimum value being that necessary to secure the entrance of a tooth into action before the preceding tooth has gone out of action—that is, the arc of contact must be greater than the pitch. The shorter the teeth are for a given pitch the stronger they are, and the less the action. Also an inspection of the cycloidal forms in fig. 48 will show that the higher the tooth the more inclined to the tangent at the pitch point will the common normal to the teeth become, increasing the force between the teeth and load on the gear bearings in comparison with the driving force. The ratio of height to pitch should therefore be as small as possible consistent with safety and smooth running, and this has led to the introduction of the stub system of gear teeth.

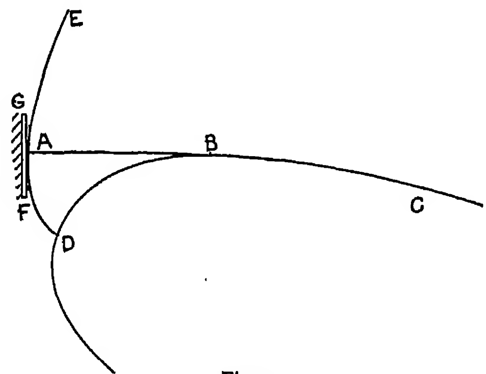


Fig. 50

Involute Teeth.—If a cord ABC wrapped round a curve (fig. 50) be supposed unwrapped (without slip), any point such as A traces out the involute DAE of the curve. The point B at which the cord leads off from the curve is the centre of curvature of the involute at the point A, and a straight line FAG at right angles to AB touches this involute. All involutes of circles will clearly be similar curves, the shape being completely fixed by the radius of the circle.

In fig. 51, ABC, DEF are circles, centres O_1 and O_2, having AD cutting O_1O_2 in G as a common tangent. The circles may be supposed connected by a cord along the common tangent, so that by turning ABC the cord is wound on to this circle and unwrapped from circle DEF, turning the circle. If ABC be fixed and O_2 moved round it, any point P on the cord traces out an involute on the part ABC, and conversely P will also trace out an involute on the part DEF. When the point P is in the position shown, these involute curves are as indicated by the broken lines, and the common normal PGA intersects O_1O_2 in the fixed point G, and therefore such curves will serve as tooth outlines. The pitch lines of the gears will be circles HGK, LGM, passing through G with centres at O_1, O_2. The path of contact will be DGA—for the point P travels along this when the pitch circles revolve, carrying the "base" circles ABC, DEF with them. The path of

contact, therefore, is a straight line making the angle α, termed the obliquity, with the tangent NG to the pitch circles.

The tooth outlines for the gears are involutes drawn to the base circles. If the centres of the base circles had been set at O_1 and O_3, the obliquity of AD would have been altered, but the tooth shape would work in the same manner, so that involute gears have the property of working correctly at different centre distances. In the design of the teeth a particular obliquity is selected, and the tooth dimensions arranged to suit this. If the gears

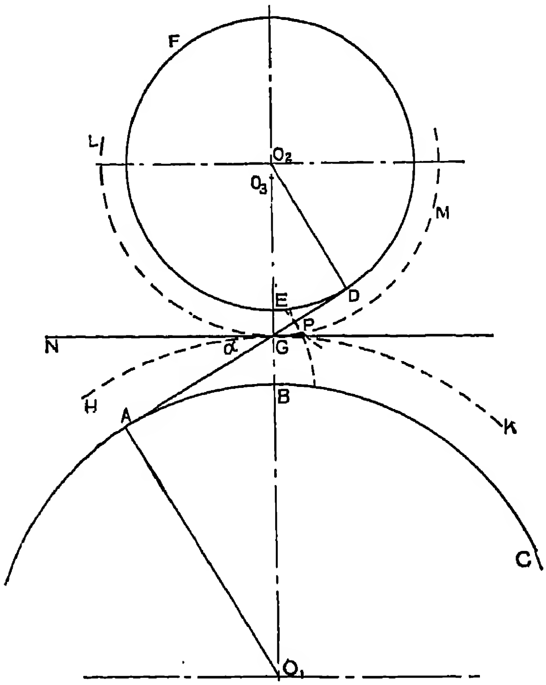


Fig. 51

are put into mesh for this particular obliquity, the tooth clearances, arc of contact, &c., are as planned. If the gear centres are farther apart the clearances are increased, and the arc of contact lessened, but the velocity ratio is still maintained constant. Thus involute teeth should be used when the centre distance is not precise, as with change wheels.

The involute shape gives in general a stronger tooth than the cycloidal, being convex throughout; but where pinions with few teeth are required interference occurs, and the teeth have to be undercut to clear the points of the teeth of the engaging gear.

Internal gears, having the base circle within them, have the points of the teeth at the start of the involute. Rack teeth (involute) are straight lines perpendicular to the path of contact. They are thus simple to produce, and this fact lies at the basis of the generation of involute teeth.

In the system of gear teeth due to Messrs. Brown & Sharpe the addendum is made to be the module or $\frac{1}{\text{diametral pitch}}$, which gives an easy method for calculating the overall diameter of the gear blank, namely, by dividing two more than the number of teeth by the diametral pitch. The clearance at the root of the teeth is made a tenth of the thickness of the tooth on the pitch line. The intended obliquity of the involute is $14\frac{1}{2}°$.

In the short or stub tooth system the obliquity is made 20°, and the addendum $\frac{1}{4}$ the circumferential pitch or $\frac{7}{8}$ module.

Helical Gearing.—Since the action in a plane perpendicular to the axis of a pair of gears is independent of that in the others, the section could be different in each plane. To secure smooth action Hooke proposed that the gears should have the teeth stepped—a form which is now employed, the gears being built up of stampings—which leads to the case where the gears have the teeth cut helically or, in order to prevent axial thrust, double helically, or " herring-bone ". Where gears are milled the helical tooth introduces the difficulty that the section perpendicular to the gear axis is different from that normal to the helix, and this again is slightly different from the tooth shape of the cutter necessary to produce it. In generated gears the same difficulty does not occur, and where the teeth are shaped the difficulty of the angle of the herring-bone also disappears. Not only do these gears give a smoother action than straight-cut gearing, owing to the spreading of small imperfections, but the tooth is also somewhat stronger.

Bevel Gearing.—When the axes of the gears meet at a finite point, the pitch surfaces are the friction cones of fig. 42, which may have axes at any angle, and may have contact corresponding to internal spur gears. A case of particular interest is where the pitch cone of one gear becomes a plane; the apex of the gears lies on the plane and the gear is termed a crown gear.

If the teeth of bevel gears be generated by a line passing through the apex, the sections at all distances from the apex will be similar, so that if the action is correct at one distance from the apex it will be correct at all distances. The gears may now be supposed to be divided up by spheres, and these turned round the axes of the gears, producing spiral gear teeth in the same manner as herring-bone spur teeth are geometrically produced.

The shape of the teeth can be set out by supposing the cone enveloping the outer ends of the teeth, or more precisely cutting the pitch cone there perpendicularly, to be developed as shown in fig. 52, drawing the teeth on it to satisfy the conditions previously found, and replacing the conical surface in position. Actually, the teeth should be set out upon the enveloping sphere, so that this method introduces an error, which is, however, small. The teeth may be produced by a tool whose edge is arranged always to pass through the apex of the gears, and is guided by a template produced as above.

Bevel gears were first produced by the figenerating process by

H. Bilgram, who constructed machines for the purpose and produced cut gears commercially. The gear blank is rolled mechanically upon a plane, its turning being controlled by steel strips attached to a former, or conical arc of the pitch angle required. The plane, or rather its extension, is the pitch surface of an imaginary crown gear with straight-sided teeth,

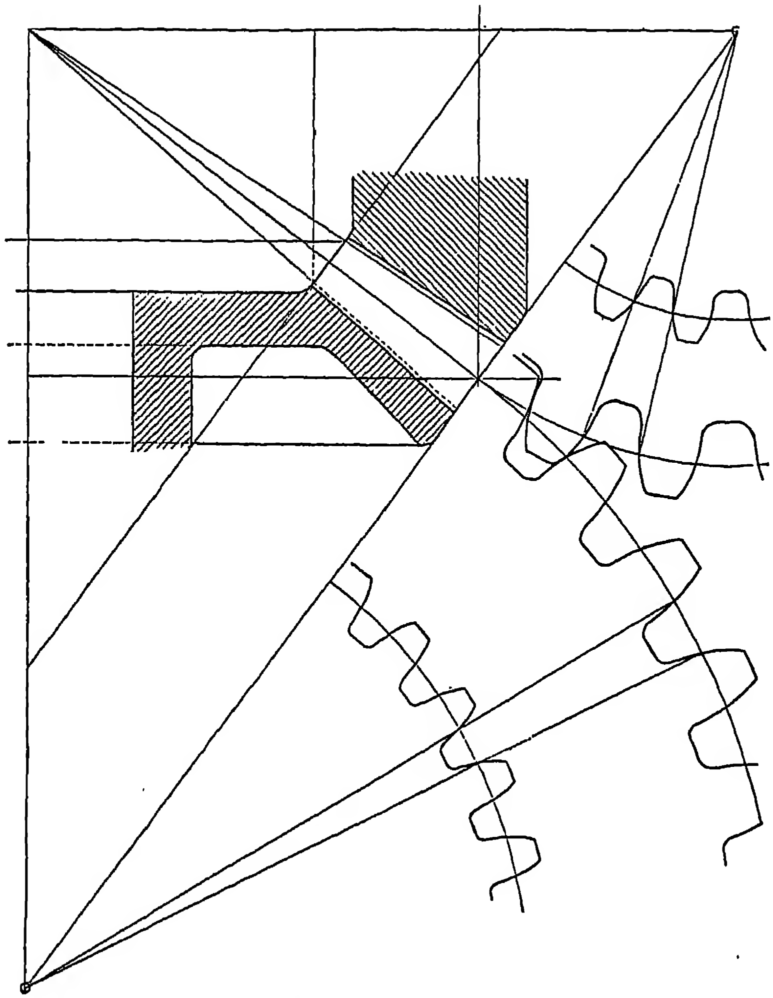


Fig. 52.—Bevel Gears, Form and Size of Teeth

which is actually swept out by the edge of a tool, the plane in which the edge moves intersecting the fixed plane in a line through the apex. The tool edge corresponds to the line FAG in fig. 50, and the plane swept out by it envelops a correct gear tooth shape, as the plane through the line of contact of the gear and perpendicular to the cutting edge must pass through the instantaneous axis of the rolling. The cutting edge, not being a mere cutting point, maintains its shape well, and as it cuts by enveloping (there

is a series of cuts so that the tooth produced is not a mathematical envelope) produces a tooth shape smooth and very nearly of the precise form. Furthermore, the gear produced not only acts correctly with the imaginary crown gear but must act correctly with a gear of any vertical angle meshing with this crown wheel, and so correctly meshing bevel gears can be produced. This machine laid the foundation of the manufacture of spur and bevel gears (straight or spiral and to mesh interchangeably) by the generating system, which produces the tooth shape in the machine itself and not through the medium of a template. The Sunderland gear planer most nearly corresponds to it in the generation of the teeth of spur gears, although the Fellows gear shaper, in which a cutter is first generated and then is used to shape the gear blank by the corresponding relative rolling motion, preceded it.

Pin Gearing.—A common type of gearing in clock work is pin gearing, the pinion consisting of pins of round section mounted in a plate or between two collars. The pitch circle of the pin gear is the circle through the centres of the pins. If the pitch circle is employed as the rolling circle for the generation of cycloidal teeth, a point on it gives a mere point as the hypocycloid, formed by rolling in the pitch circle of the pin gear, and epicycloids on the other gear. The points being replaced by pins, the epicycloids will be replaced by the envelopes of circles (of pin diameter) moving along the epicycloids, that is, by curves parallel to the epicycloids, as shown in fig. 53. The pin gear therefore has no face to the teeth and the mating gear no flanks, and if the pin gear be the driven, the action takes place during the arc of recess, which is favourable from the point of view of friction.

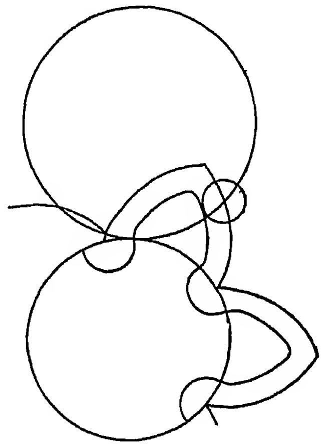


Fig. 53

Spur Gear Efficiency.—The efficiency of well-cut spur gearing is very high, lying between 0·96 and 0·99 not including journal friction. Spur and bevel gearing is often referred to as "sliding" gearing to distinguish from friction gearing which rolls, although the action of the tooth surfaces consists of a roll combined with a slide, the relative rolling being the relative angular velocity, and the sliding this quantity multiplied by the distance of the point of contact from the pitch point.

Gears in Trains.—The velocity ratio communicated from a shaft having a gear A of m_1 teeth working with a gear B of m_2 teeth on another shaft is $-\dfrac{m_1}{m_2}$, the negative sign implying that the shaft B revolves in the opposite direction to that of the shaft A, the gears being external. If one gear be internal, the ratio is $+\dfrac{m_1}{m_2}$. Thus if A revolve at n_1 revolutions per

minute and B at n_2, then $n_2 = -\frac{m_1}{m_2}n_1$ for external gears and $n_2 = +\frac{m_1}{m_2}n_1$ if one gear be internal. If an additional gear, termed an idler, of any number of teeth be interposed between external gears A and B, the only effect is to render positive the direction of rotation of gear B. The velocity ratios obtainable are limited by the fact that m_1 and m_2 must be whole numbers, and also by the large size of the gearing if the velocity ratio is high and the pitch that requisite from considerations of strength; but greater flexibility is obtained by compounding the gearing. If, in fig. 54, the gear A, having m_1 teeth, drive B of m_2 teeth, and to the shaft to which B is fixed the gear C having m_3 teeth be also fixed and arranged to drive the gear D of m_4 teeth, the velocity ratio between A and D is compounded of that between A and B with that between C and D. If A make one turn the system BC makes $-\frac{m_1}{m_2}$ of one turn, corresponding to which D makes $\left(-\frac{m_1}{m_2}\right) \times \left(-\frac{m_3}{m_4}\right)$ of a turn, so that the ratio is $+\frac{m_1 m_3}{m_2 m_4}$. With the numbers of teeth given in the figure the ratio is $\frac{18}{64} \times \frac{20}{45}$ or $\frac{1}{8}$. If the arrangement is to transmit force the pitch of the C–D teeth should in the example be made greater than that of the A–B teeth. The compounding may be extended, employing more axes.

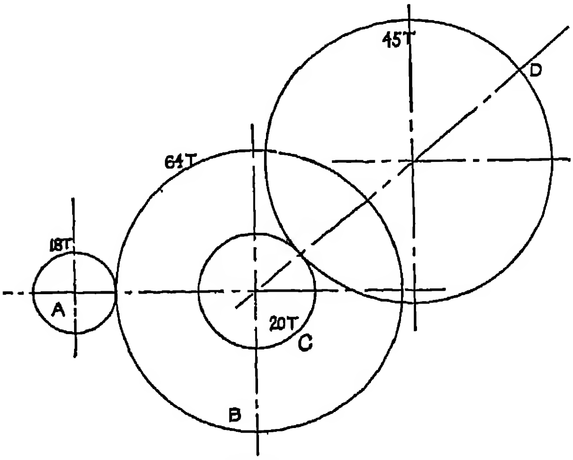


Fig. 54

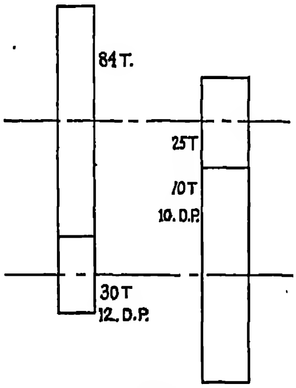


Fig. 55

Reverted Trains.—If the axis of the final wheel is the same as that of the initial wheel the train is termed "reverted". The centre distance for each pair of gears should be the same, though this is not always adhered to, as a small difference may be covered by setting the gears more or less deeply into mesh. The gearing of a lathe headstock (fig. 55) forms a reverted train. In the example given the velocity ratio is $\frac{30}{84} \times \frac{25}{70} = \frac{1}{7{\cdot}84}$,

so that if, with the belt on a particular step of the cone, the spindle makes 250 r.p.m. without the back gear, it will make $\frac{250}{7\cdot84} = 32$ r.p.m., nearly, with the back gear in. The centre distance with the pitches given is $4\frac{3}{4}$ in.

Epicyclic Gearing.—In fig. 56 is shown a reverted train of gears, the axes being carried on an arm E, the velocity ratio being $\frac{m_1}{m_2}\frac{m_3}{m_4}$, so that if a single turn be given to the gear A when the arm E is fixed, the gear D makes $\frac{m_1m_3}{m_2m_4}$ turns. If, instead of A being the driver, it is fixed and the arm E driven round, the gear train is termed epicyclic, A being commonly termed the sun and B, C planet wheels. Practical epicyclic trains are reverted and have two or more sets of planet wheels for balance. Taking A to have made one turn, it can be reduced to rest by giving to the whole system one complete turn in the negative direction, so that the final result is that A makes no turns or is at rest, while D makes $\frac{m_1m_3}{m_2m_4} - 1$ turns.

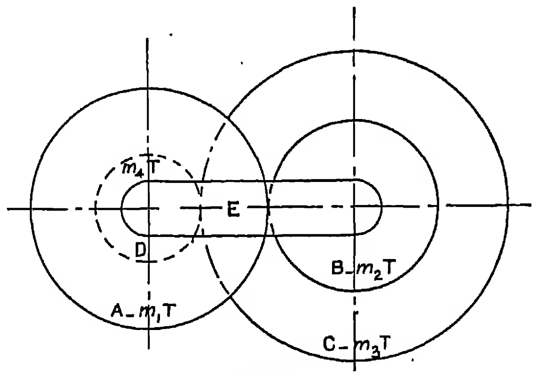


Fig. 56

In some cases both A and the arm E are forced to rotate—suppose A at N_1 and E at N_0 r.p.m., so that the gear D receives motion from two sources. The rotations can be tabulated as follows:

Gear A.	Arm E.	Gear D.
1	0	$\frac{m_1m_3}{m_2m_4}$
-1	-1	-1
0	1	$1 - \frac{m_1m_3}{m_2m_4}$
N_1	N_0	$N_1\frac{m_1m_3}{m_2m_4} + N_0\left(1 - \frac{m_1m_3}{m_2m_4}\right)$

The first line being the case where the arm E is fixed, the second representing the negative turn of the whole of the mechanism locked together, and the third obtained by the addition of the preceding two lines.

Multiplying the first line of revolutions by N_1 and the last by N_0 we get the set given in the last line, the r.p.m. of D being $N_1\frac{m_1m_3}{m_2m_4} + N_0\left(1 - \frac{m_1m_3}{m_2m_4}\right)$.

By epicyclic gearing high velocity ratios can be obtained, when the transmission of the high forces which accompany high velocity ratios in power transmission requires careful consideration. By employing more than one sun wheel, convenient change-speed systems can be obtained, operated by applying brakes to the sun wheels.

Bevel gears afford a convenient form of epicyclic gearing for the differentials of auto-car axles. In fig. 57 is shown a back-axle drive, the power being

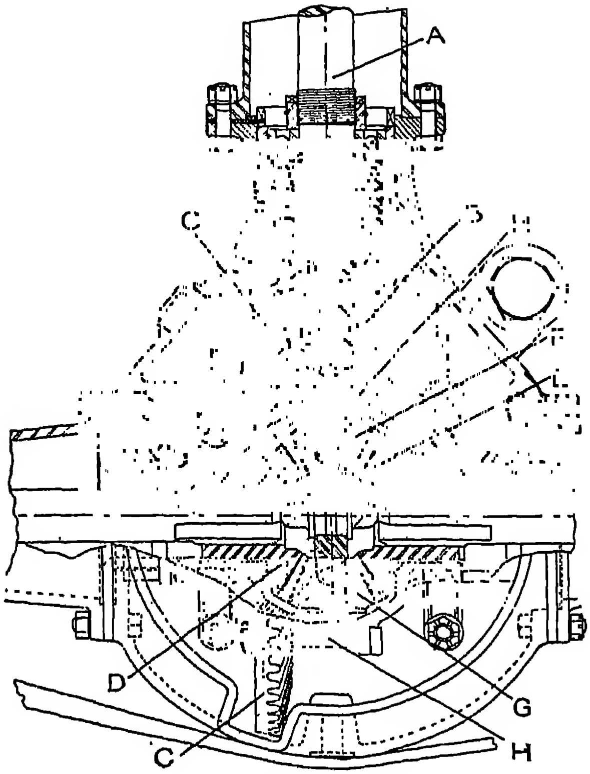


Fig. 57

delivered by the shaft A carrying the bevel pinion B, which drives the bevel gear C carrying the planet wheels of the epicyclic, consisting of mitre wheels D and E fixed to the shafts of the axle, and planet mitre wheels F and G—two for balance—loose upon journals fixed in the differential casing H carried by the bevel gear C. In straight driving the whole gear C, the casing and differentials and halves of the back axle revolve as a single body, but in turning a corner the differential allows the wheels to revolve at different speeds, the average being that of the casing.

If F be fixed and one positive turn given to D, the gear E makes one turn in the negative direction; superposing one negative turn on the system shows that if one wheel D be fixed the other E will rotate at twice the speed of the driving bevel C with the cage.

Gears with Non-intersecting Axes.—When the axes do not intersect, the gears are termed screw gears, worms and worm wheels, or hyperbolic gears.

Screw Gears.—Screw gears have the teeth cut in helices or screws on right circular blanks. The axes of screw gearing may be inclined at any angle, and the velocity ratio is only partially controlled by their diameters. The customary drive from the crank shaft to cam shaft of gas-engines, and worms meshing with straight-cut spur gears—the angle of the worm shaft being inclined at the pitch angle of the worm to the plane of the gear—are examples of such gearing.

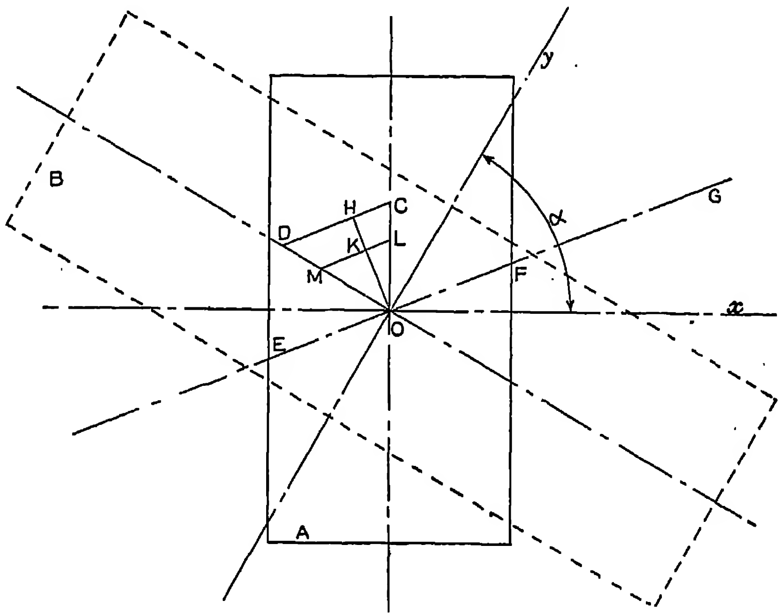


Fig. 58

Let Ox and O$_1y$ be the axes of screw gears A and B at angle α, the angular velocities being ω_1 and ω_2 respectively. The pitch cylinders of the gears touch at O$_2$, and in fig. 58, which is a view taken along the shortest distance between the axes, O, O$_1$ and O$_2$ all appear at O. The gear A is shown in outline, and B, supposed removed, shown by broken lines; let them have radii r_1 and r_2. Then considering a small movement, the point on gear A in contact at O will move to C, where OC $= r_1\omega_1\delta t$, and the corresponding point on gear B will move to D, where OD $= r_2\omega_2\delta t$, and hence the direction of the teeth in contact must be CD, so that the teeth at O lie in the parallel direction to EFG.

The normal pitch p_0 of the gearing will be measured along OH perpendicular to CD, and is the same for the two gears. Marking off OK $= p$ along OH, and drawing LKM parallel to CHD, the circumferential pitches OL and OM of the gears are obtained. The pitch circles must contain OL

and OM integrally. The gear ratio $\left(\frac{\omega_1}{\omega_2}\right)$ will be inversely as the number of teeth on the gears, which gives the circular pitches corresponding to selected pitch diameters.

The contact of screw gearing is at most point contact, so that the force transmitted between the gears is small relatively to that transmitted by spur and bevel gearing which have line contact.

Worms and Worm Wheels.—A worm is merely a portion of a screw thread, and the gear which meshes with it, the axes being usually at right angles, is termed the worm wheel. As almost all worm wheels are cut, and have their axes perpendicular to the worm axis, that is taken to be

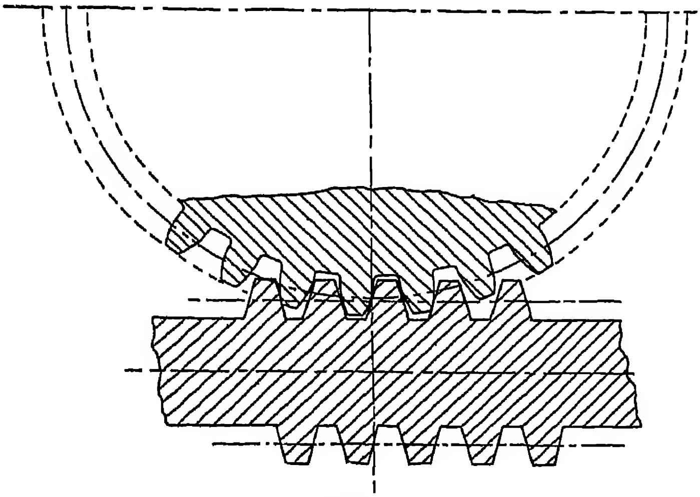


Fig. 59

the case in this discussion. The central section of a worm and worm wheel is shown in fig. 59. As the worm rotates its section will remain of the same shape, but the teeth of the section will travel axially, simulating the movement of a rack. If then the central section of the worm wheel be formed in accordance with the principles of spur gearing for transmitting uniform velocity ratio, it will be of the correct shape to gear with the worm. Thus if the worm have a straight-sided section, as in fig. 59, the central section of the worm-wheel teeth will be involutes. The simplest mode of regarding this is that the central section of the worm-wheel teeth is to be formed by rolling the worm wheel upon the central section of the worm regarded as a rack, which will have the same result as is produced by a hob of the worm shape rotating in contact with the worm-wheel blank driven to rotate with the desired angular velocity ratio. Thus the central section of the worm wheel is generated. If a parallel section be considered, it can only be cut by the parts of the corresponding section of the hob, which is the same (except top and bottom clearances) as the worm, and which sections —of whatever shape—may be viewed as a rack travelling with the same

velocity as the central section. Thus this section of the worm wheel is by the generation produced correctly.

At every section, therefore, of worm and worm wheel taken perpendicular to the axis of the worm wheel there is always one point of contact between corresponding teeth. The sequence of these points of contact for the various sections therefore constitutes a line of contact, so that a worm and generated (hobbed) worm wheel have line contact across each tooth in action.

From this point of view the shape of the worm section and whether it has one or more threads make no difference—each tooth has line contact maintained across it. The shape of the central section is of importance from two points of view, that of interference and that of efficiency. Geometrical interference always takes place when the face of the worm wheel is wide, but this is still advisable from the point of view of load carrying. Further interference, or removal of contact surface, may take place, owing to the method of hobbing being to feed the worm for its cut radially towards the worm-wheel axis: this is avoided if the worm axis be correctly located initially, and, in addition to the amount of rotation given to the worm wheel corresponding to the rotation of the worm, an axial feed is given to the worm and the rate of rotation of the worm wheel correspondingly increased. The hob shape must then be tapered to give a suitable cut per tooth. A shape of tooth like a standard involute rack gives satisfactory results, but various shapes have been advocated.

The action between the surfaces in contact consists of slide, roll, and spin about the common normal, the amounts being readily found by the methods of solid geometry, and depend upon the distance of the point considered from the instantaneous axis of the relative motion and of the direction of the common normal.

Efficiencies.—The efficiency of a screw and nut, with which mechanism the worm and worm wheel is closely related, has been considered earlier (p. 13), and it is shown that the main factor is the average angle of the thread helix, the efficiency being low for small angles and high for angles which are less than 45° by half the angle of friction. The interaction of the surfaces in worms and worm wheels facilitates lubrication, and the coefficient of friction is low when the speed is sufficiently high. The shape of the worm thread affects the efficiency in the same way as it does that of a screw nut. When the helical angle is low, as is the case with single-thread worms, the efficiency is low, and the reversed efficiency, i.e. for the worm wheel to drive the worm, is zero. This is convenient in many cases—such as instruments, hoists, and steering gears—as the worm wheel is locked by the friction, while the velocity ratio obtained is high. Worms and worm wheels being at first used to give high velocity ratios (for adjustments and control of large forces), the involved low efficiency of the mechanism led to a widespread but erroneous belief that it was inherently low, which delayed the adoption of the device for power transmission. Design following the principles of screw efficiency to worms and worm wheels for speed reduction from electric motors at once gave efficiencies of about 80 per cent.

Modern finely made gearing, with multiple-thread worms, gives efficiencies of 90 to 97 per cent, which compares not unfavourably with spur and bevel gear efficiencies. The velocity ratio (number of teeth in worm wheel/number of threads on worm) obtained by the use of multiple threads instead of single threads (for the purpose of efficiency) is not high, 5 to 8 being common values.

Hindley Worm.—A type of worm and worm wheel in which the worm is curved in outline to suit the worm wheel was invented by Hindley, and is known by his name.

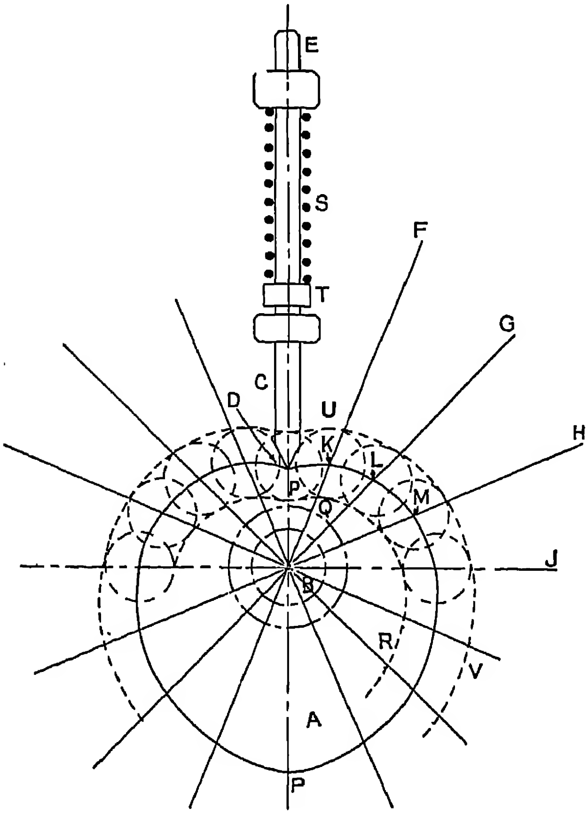


Fig. 60

The surfaces of the Hindley worm and worm wheel lie as a whole much closer together than those of parallel worms and their wheels, and the contact is of a higher order, so that much higher loads can be carried. The efficiencies of well-designed Hindley gears are over 90 per cent.

For high power transmission in comparison with the size of unit, worms are made of hardened steel, finished to a very fine surface, and the wheels of phosphor bronze.

Teeth may be formed upon pitch surfaces corresponding to friction gearing of any shape, such as those shown in figs. 40 and 41, the latter being occasionally used for driving reciprocating parts to give a more uniform motion than is given by a simple crank. The teeth are set out in accordance with general principles, or the shape of those on one blank may be arbitrarily chosen and the shape of the teeth on the other blank determined by the method of envelopes.

Cams.—Irregular motion is usually transmitted by means of cams, which consist of interacting surfaces which may be separated by a roller to prevent wear from affecting the shape of the surface. In fig. 60 is shown a cam A, driven by the shaft B, operating a piece C, arranged to slide so that its pointed extremity D moves in a straight line through the

centre of the shaft B, and giving to C a uniform velocity to and fro, when the cam revolves uniformly. The cam shape can be easily drawn, as it is the envelope of the point D in its movement relatively to the cam, which may be supposed at rest, the construction being to set off a series of relative positions BF, BG, BH... of the line BDCE and mark off on them BK, BL, BM..., the corresponding distances of the point D of the slider from B. For the uniform movement of D the distances BD, BK, BM increase by equal intervals, the angles between the lines BE, BF, BG, BH, &c., being equal. After the curve DKLM reaches the line DB at P, the motion is supposed to reverse, and the cam is symmetrical. This cam is termed a heart cam from its shape.

The contact of the point D with the cam outline is unmechanical, and the point should be rounded, or a pin and roller P as sketched in broken lines employed. The outline of the cam is now the envelope of the relative movement of the roller—constructed by drawing the roller circles (broken lines) for the points D, K, L, M, &c., giving the inner (broken line) shape QR. The roller rolling on the cam transfers the sliding motion to that between roller and roller pin.

The driven part or "follower" C must maintain the contact of its point or roller with the cam during the motion, otherwise the specified motion will not be attained. In the illustration a spring S acts on a collar T so as to force the slider C to maintain contact, the element of the mechanism being then force-closed. It may, however, be closed by a second part, UV, of the cam, having the shape of the envelope of the other side of the roller positions, and usually taking the shape of a groove in a plate.

Suppose that contact is not maintained by the second envelope method, and that the part C is now returning following the part of the cam beyond P; if the cam speed is high enough the follower will leave the cam, in which case the tension in the spring S must be increased. If T be the kinetic energy of the driven mechanism, and V (the friction being taken as negligible) is potential energy, the condition for maintenance of contact (being the existence of no force between cam and follower) is that $\frac{dV}{dx}$ is always greater than $\frac{d}{dt}\frac{dT}{dv}$, where x is the position of the follower and v its corresponding velocity. The problem of contact maintenance is one of acceleration, which can be found by the methods previously given (Vol. II, p. 120).

In the particular cam drawn it will be noticed that there is a sudden change of velocity at the points D and P of the cam, the velocity suddenly reversing and the acceleration infinite; thus the follower must leave the cam at P and the force between them at D is very high, so that such a cam is unmechanical, and, except for very slow speeds, its shape at D and P must be altered.

The force on the cam roller (except at position D and P) is inclined to the line BE; thus at position BLG the force makes an angle with the line

BLG and therefore with the line BE in actual position, so that there is a side force on the follower.

This action can be reduced to insignificance by the employment of a lever, as in fig. 61. The cam shape is now determined as the envelope of the relative positions of the lever.

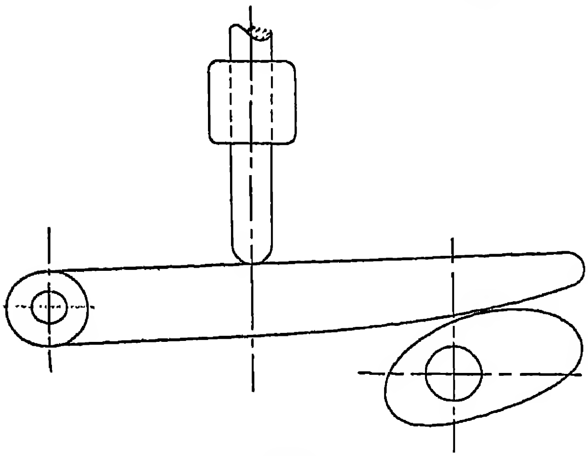


Fig. 61

The above cams are edge or cylindrical cams, the working surface being cylindrical with its generators parallel to the shaft of the cam: cams grooved on the perpendicular surface are termed face cams.

Ratchets and Pawls.—Intermittent motion may be produced by ratchets and checked by pawls, as shown in fig. 62, in which the ratchet wheel is driven by the concentrically pivoted arm B, to which is hinged the ratchet C. The teeth are to be cut at such an angle that the ratchet is held in gear by any thrust between itself and the teeth. At E is shown a detent which prevents the wheel returning while the ratchet is moving back. As shown, the lever B is operated by the sliding bar F, which pushes it forward to a fixed limit. The arm B carrying the ratchet returns (by its own weight) in contact with F, until it comes into contact with the point G of the screw H. By adjusting the screw H the arm B is allowed to fall back so that the ratchet slips over one, two, or more teeth of the ratchet wheel, the movement of the ratchet wheel corresponding to a fixed movement of the operating rod F, being therefore variable.

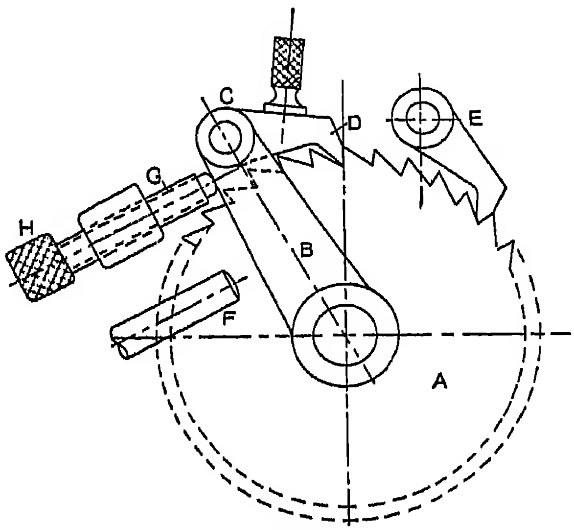


Fig. 62

In fig. 63 is shown a ratchet wheel with symmetrical teeth, and a ratchet which can be reversed from the position shown in full lines to that shown in the broken lines. The ratchet slips out of and over the teeth when it

moves in the direction of its pivot, owing to the angle on those edges of the ratchet.

Multiple Ratchets.—The movement in these ratchets is an integral number of teeth, but with the same pitch half this movement can be produced by using a pair of ratchets on the arm, as is shown in fig. 64, and by the use of more ratchets the division of the tooth space can be

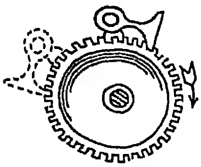


Fig. 63

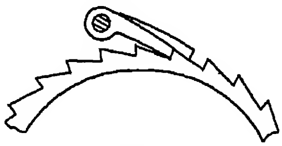


Fig. 64

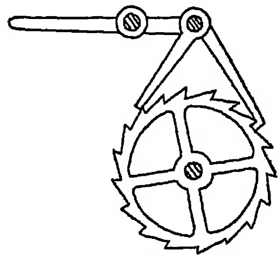


Fig. 65

carried farther. In fig. 64, as the arm returns from the position shown, at half a tooth space movement, the ratchet at present not working drops over a tooth, and on the forward movement becomes the working ratchet. The certainty of action and strength of tooth can be thus retained.

A pair of ratchets are also employed when the ratchet wheel is to move at each movement of the operating arm, an arrangement being shown in fig. 65.

Masked Ratchet Wheel. — The reverse problem, one movement of the ratchet wheel corresponding to two of the operating gear, is solved by the employment of a masked ratchet wheel, shown in fig. 66. Behind the operative ratchet wheel is the masking ratchet, which has teeth of alternate depth, the ratchet operating upon both wheels. In the position first shown the ratchet is engaged in the tooth of the wheel only, the depth of this tooth not being sufficient for the ratchet to engage the tooth of the primary ratchet wheel. The stroke of the ratchet causes the masking ratchet wheel only to move, and on its return the ratchet falls into deeper space on the masking wheel, which is cut so low that it also engages the tooth of primary ratchet wheel.

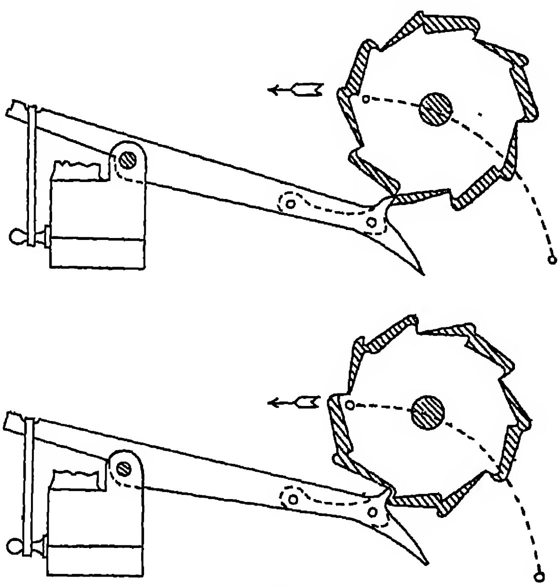
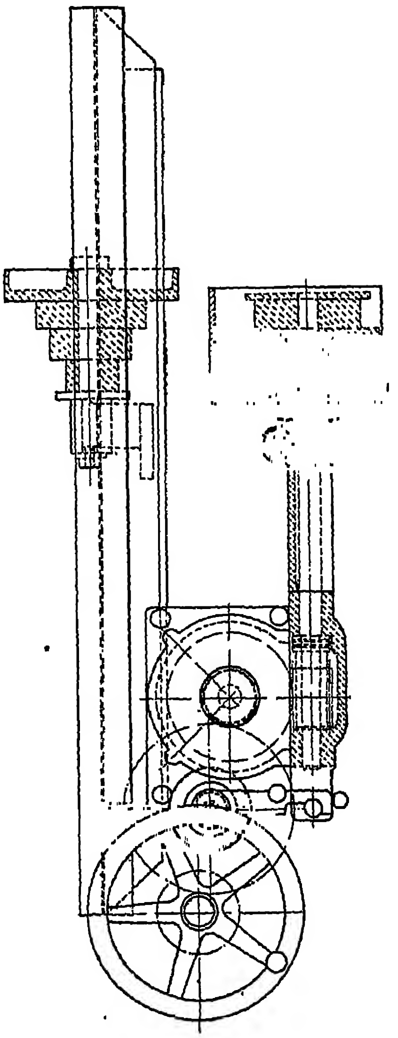


Fig. 66

Throw-out Mechanisms.—If a ratchet be required to feed to a certain position the teeth would not be cut beyond that place, but where the position is variable it is customary to fit an adjustable shield which can be moved round the ratchet wheel by hand and clamped in any position from it.

Precision of throw-out is important in many machines; one of the most effective mechanisms, the dropping worm, is shown in fig. 67. Upon the knife edges of the trip passing one another the worm drops out of gear with the worm wheel, rotating with its shaft and bearings about the supporting pin. In throw-



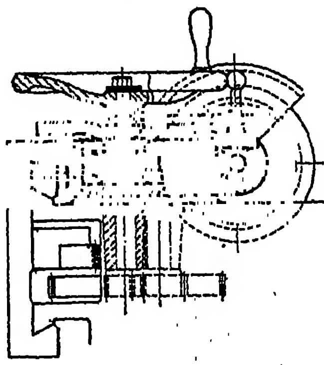


Fig. 67

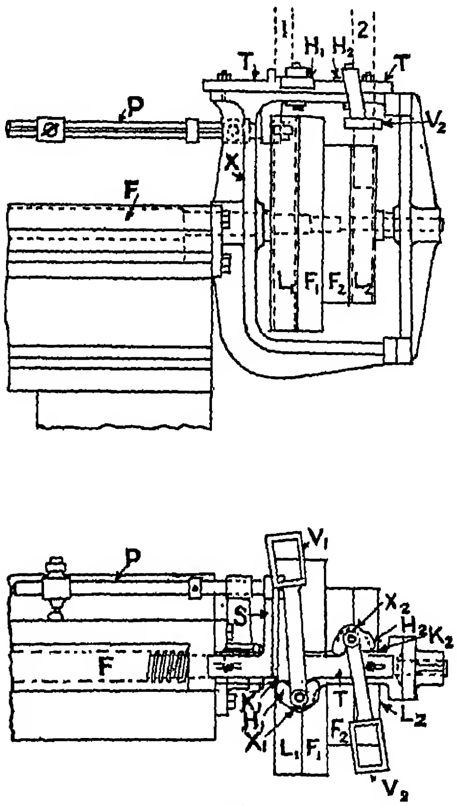


Fig. 68

outs all the forces on the moving part should tend to accelerate the action.

Reversing Mechanism.—For reversion of motion many arrangements are in use, depending upon the circumstances. In fig. 68 is shown the reversing gear of a side-planing machine, in which the difference between the speeds of cutting and reverse is obtained by belting and pulleys. F_1 and F_2 are fast pulleys, and L_1 and L_2 are loose pulleys. The slipper rod P is operated by stops on the moving slide, and moves a sliding piece T, which operates the belt slippers V by means of rounded cam plates. The fast pulleys are both fastened directly on to the screw driving the slide. As the belts must never be on the fast pulleys simultaneously, the driving belt is shifted first, and the energy of the table then moves the other belt over. In the reversal of heavy slides the acceleration presents the most serious problem, springs and other energy containers being employed to obtain it.

The Quadric Crank Chain.—In fig. 6, Vol. II, p. 89, is a linkage consisting of four bars connected by four pin joints (having parallel axes), and it is proved that there is one degree of relative freedom. In this mechanism, having four turning pairs and termed a quadric crank chain, one pair may be replaced by a sliding pair (equivalent to taking the pin joint to infinity), when it is called a slider crank chain; or a further turning pair may be replaced by a second sliding pair, it being then termed the double-slider crank chain. These chains form fundamental mechanisms in many machines.

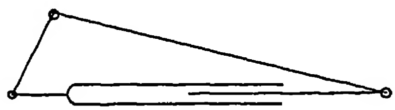


Fig. 69

One link in each chain is fixed in a machine, the fixing of the different links, although introducing no difference into the relative movement of the parts, produces machines suitable for different purposes and of very different appearance. The change of the link which is to be fixed is termed inversion.

The relative lengths of the links also make important differences. For example, the side rod of a locomotive with the two cranks and the frame is a quadric crank chain, having two long equal links and two short equal links. If one short link be made to revolve, the second short link is driven to revolve. In a beam engine, the frame of the machine, the beam, connecting rod, and crank form a quadric crank chain, in which the lengths of the links are such that a rocking movement of the beam is combined with a rotating movement of the crank, while in a Watt parallel motion both the beam and the link merely rock.

The Slider Crank Chain.—Since the slider crank chain (fig. 69) consists of four links connected by three turning and one sliding pair there is the choice whether a link containing a turning and a sliding pair be fixed, or whether a link containing two turning pairs be fixed. As was the case with the quadric crank chain, in which there was not this choice, there are variations introduced by the relative shapes of the links. Of the first class the direct-acting steam-engine (fig. 15, Vol. II, p. 93) is the most familiar example, the fixed link carrying the cylinder and main bearing,

the piston and rod having a slide and gudgeon pin, the connecting rod having two bearings, and the crank connecting the crank-pin and main journals being the other links. The engine may be désaxé* without affecting the classification. The simplest form of quick-return crank-driven mechanism, used in small shapers and planers, in which the connecting rod is connected to the main slide, and some fly-presses, are further examples.

If a link connecting two turning pairs be fixed, the machines resulting have a different appearance. A distinction may be drawn between those in which the longer link, corresponding to the connecting rod in fig. 15, Vol. II, p. 93, is fixed ("crank and slotted lever mechanism"), and those in which the shorter, or crank in the same figure is fixed (pin and slot mechanism). Of the former class the obsolescent oscillating cylinder engine and a quick-return motion employed on many shapers are examples.

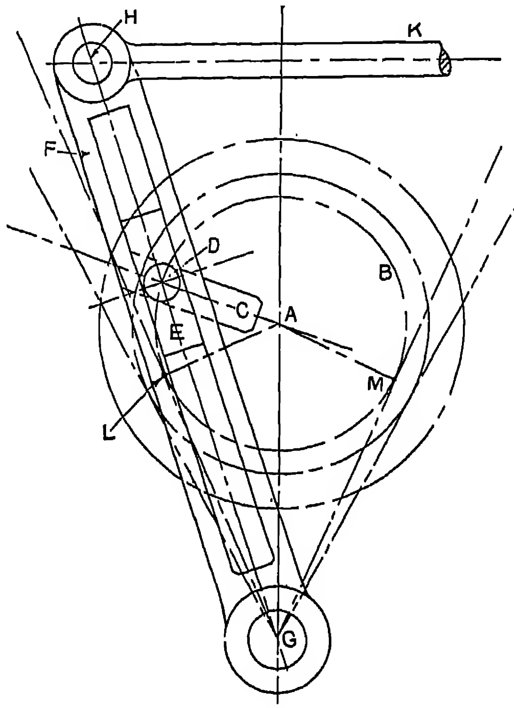


Fig. 70

Quick-return Motion.—The latter is shown diagrammatically in fig. 70, A being the driving shaft carrying the disc B, with a slot C along which the pin D is adjustable, so that the stroke of the machine can be varied—in a well-designed machine this can be done while the machine is running. The pin D bears in the sliding block E, which works in the slide in the swinging arm F, which is pivoted at G. From a pin at H on the arm F the connecting rod HK runs to the ram of the machine. The arm and block are usually symmetrical, as in the figure. The limits of the stroke occur when the centre line of F touches the circle in which the centre of the pin D travels. When the pin is in the position sketched the tangents are GL and GM, so that the ratio of the time of the cutting to return stroke is that of the arcs into which L and M divide the circle. This varies with the stroke, being more unfavourable as the stroke becomes less.

Whitworth Quick-return.—The second of the above classes is represented by the Whitworth quick-return motion for shaping machines, shown

* An engine is "désaxé" or "offset" when the centre line of the shaft does not lie in the plane containing the centre lines of the cylinders.

in fig. 71. The disc B, whose centre is at A, is driven by a pinion meshing with teeth on the edge of the disc, and carries a pin D which bears in a block E, sliding in the link F which is pivoted at G. This link has a slot C formed in it, along which the pin H can be adjusted (for the purpose of varying the stroke), and a connecting rod communicates the motion from the pin H to a pin K on the ram of the shaper. The line of motion of K should pass through the point G, and is shown thus, the line cutting the circle in which the axis of the pin D moves in L and M. The ratio of the times of cutting and return is that of the arcs into which L and M divide the circle, and is independent of the length of the stroke. For constructional reasons the wheel B runs on the large shaft indicated by the broken-line circle at N.

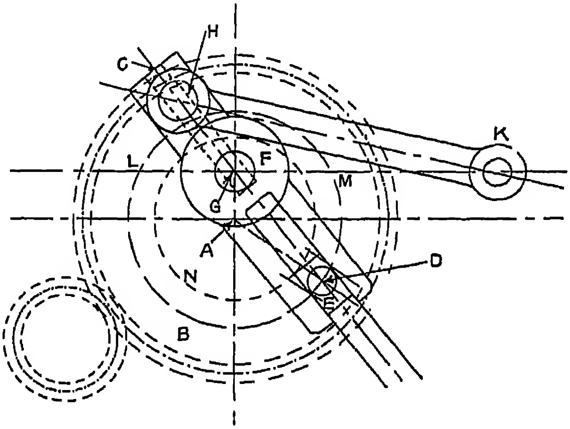


Fig. 71

Double Slider Crank Chain.—When there are two sliding and two turning pairs they may be arranged so that like pairs are adjacent or alternate. The Oldham coupling (fig. 36) is an example of this class.

Velocity in a Machine.—Since the configuration of a machine is determined when the position of one link (the frame being considered as fixed) is given, if the velocity of one link or point is known, the velocity of any point in the machine can be found—both in magnitude and direction. Also, for the same reason, if the acceleration of one link or point be also known, that of any point can be determined.

Relative Velocities.—A useful method is that of relative velocities, treated graphically, taking into account the fact that the velocity of a point B of a rigid link relative to another point A of the link is perpendicular to the line AB, and is of the amount ABω or AB $\frac{d\theta}{dt}$ where θ is the angle which a fixed line in the link makes with a fixed [1]

If v_{sa} and v_{sb} be the velocities of the p
(as indicated by the suffix s) the velocity v_a
by drawing the vector diagram fig. 72, II,

and sa to represent v_{sa}, so that as represents v_{sa} reversed, and hence the velocity of b relatively to a is ab and must be perpendicular to AB. This fact enables the determination desired to be effected. Thus if the directions of motion of A and B be known and the magnitude of the velocity of one,

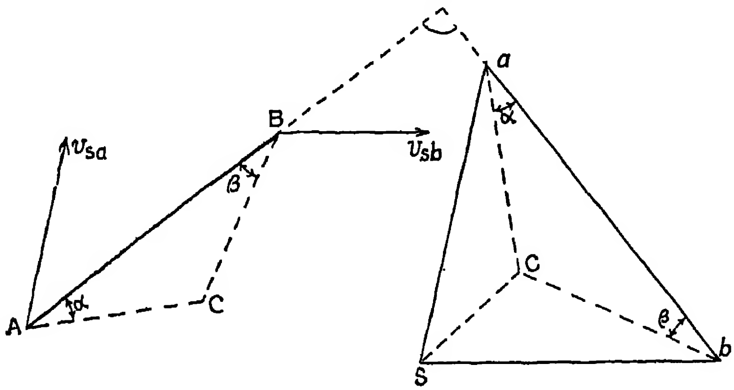


Fig. 72, I   Fig. 72, II

suppose v_{sa}, then the magnitude of the other can be found—by drawing ab perpendicular to AB to meet the line sb.

The angular velocity $\left(\omega \text{ or } \dfrac{d\theta}{dt}\right)$ of the body AB being the velocity of B relative to A divided by AB is $\dfrac{ab}{\text{AB}}$.

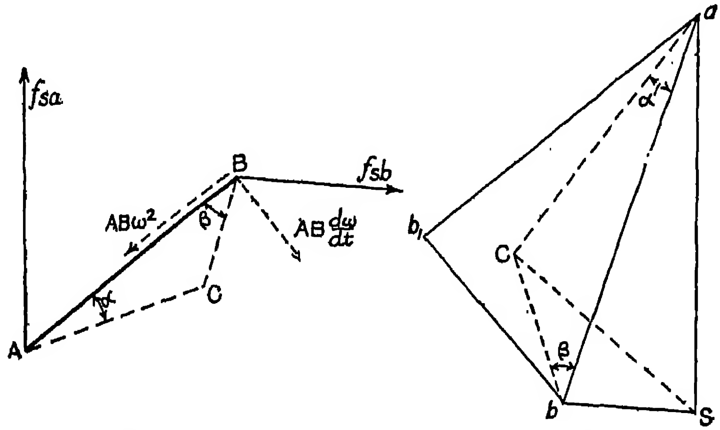


Fig. 73, I   Fig. 73, II

If the velocity of any point C of the body AB be required, since v_{ac} is perpendicular to CA and v_{bc} to CB, by drawing ac and bc perpendicular respectively to AC and BC, their magnitudes are thus determined, and the velocity of C in space, v_{sc}, is then obtained by joining cs. By the construction the triangles abc and ABC are similar. Frequently it is a point

C upon AB whose velocity is required, when the point c falls upon ab dividing it proportionally to the parts of AB.

Accelerations.—In a similar manner the acceleration of a point in a machine may be determined graphically, the main difference being that the acceleration of B with respect to A (fig. 73, I), is composed of $AB\omega^2$ radially from B towards A and $AB\frac{d\omega}{dt}$ perpendicularly to AB. The value of ω^2 can be found by the method of relative velocities (or by that of the instantaneous centre), so that the radial relative acceleration is known. The tangential acceleration is finally dependent upon that given to the primary link of the machine, whose acceleration is frequently zero.

In the vector diagram (fig. 73, II) sa is drawn to represent the acceleration f_{sa} of A in space, and sb to represent that of B. From a, ab_1 is drawn representing $AB\omega^2$, and from b_1 the line b_1b is drawn perpendicularly to ab to represent the tangential acceleration. The figure must close, affording the means of determination required.

The total acceleration of b relative to a is ba. If another point C of the body AB be under consideration, its acceleration relative to A is to that of B in the ratio of AC to AB, and makes with AC the same angle as the relative acceleration of B makes with AB. Hence making angle bac equal to angle BAC, and similarly angle abc equal to angle ABC, the point c is determined. Its acceleration in space is then sc.

As an example of the method the case of the quadric cycle chain ABCD may be considered, taking the configuration shown in fig. 74, I, A and D being fixed centres. The link AB having an angular velocity ω and an angular acceleration $\frac{d\omega}{dt}$, the angular velocity Ω and acceleration $\frac{d\Omega}{dt}$ of the link DC are required.

The velocity diagram (fig. 74, II) is constructed by drawing sb to represent $AB\omega$, perpendicular to AB, and sc in the direction of the velocity of C, i.e. perpendicular to CD, and then drawing bc perpendicular to BC to meet it, so that $sc = CD\Omega$ and $bc = BC\omega_1$, where ω_1 is the angular velocity of link BC.

The acceleration diagram (fig. 74, III) is constructed by drawing sb_1 to represent $AB\omega^2$ parallel to AB, and b_1b to represent the given value of $AB\frac{d\omega}{dt}$. Then drawing bc_1 to represent $BC\omega_1^2$ parallel to CB, when a line c_1c through c_1 perpendicular to BC will contain the point c. But if sc_2 be drawn to represent $CD\Omega^2$ (which is known from the velocity diagram) parallel to CD and then a line c_2c through c_2 perpendicular to CD will also pass through c, and hence c is determined. Then cc_2 will represent $CD\frac{d\Omega}{dt}$ and cc_1 the value of $BC\,\frac{d\omega_1}{dt}$.

In the figure the lengths of AB, BC, CD, and DA are 6 in., 16 in., 12 in., and 18 in. respectively, the link CD oscillating while AB rotates. The

link AB is taken as having an angular velocity of 100 r.p.m., and the angle BAD as 60° at the instant considered. The full lines in fig. 74, III are drawn supposing B to have an acceleration of $\frac{1}{2}g$, and the broken lines added giving the case when AB rotates uniformly. The values of the

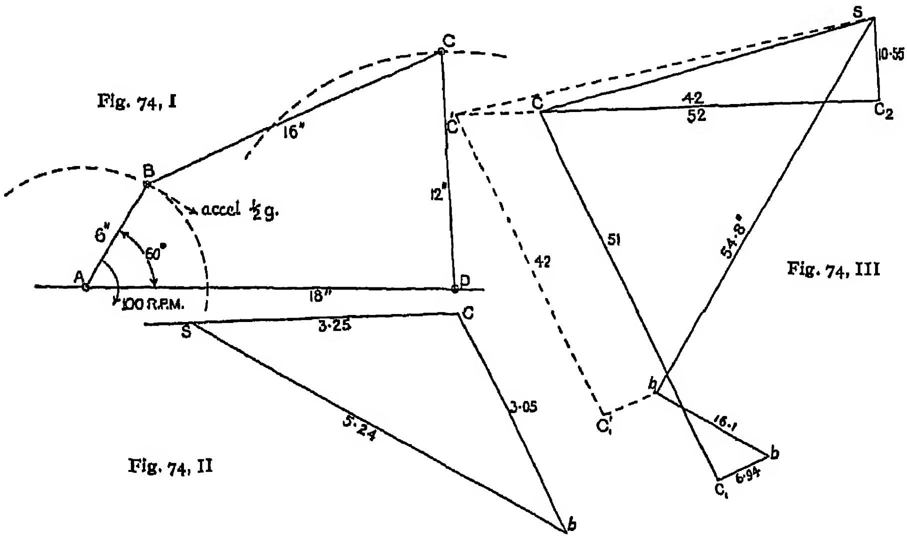


Fig. 74, I

Fig. 74, II

Fig. 74, III

velocities and accelerations are placed against the lines of the vector diagrams: the direction of c_2c shows that CD is being retarded.

The Pantagraph.—Machinery in general is built up of parts connected by lower and by higher pairs, and can be investigated kinematically by the method given above. Many machines are worthy of investigation, and attention can only be directed to a few of these more complicated examples. In fig. 75 is shown a device for making reduced (or enlarged) copies of drawings, templates, cams, &c., which is termed a pantograph. Geometrically it consists of a jointed parallelogram of bars ABCD, with one side BC produced to carry a tracing point at P, the whole being (as drawn) pivoted to the drawing board at A. If a pencil be carried at Q, where the line AP cuts CD, it will make a reduced drawing of any curve round which the tracing point P is carried. For however the shape of the parallelogram ABCD changes (by the movement of P to and from A), since $\dfrac{PC}{QC} = \dfrac{PB}{AB}$ the point Q lies on the line AP and divides it so that $AQ = \dfrac{BC}{BP}AP$. As the construction

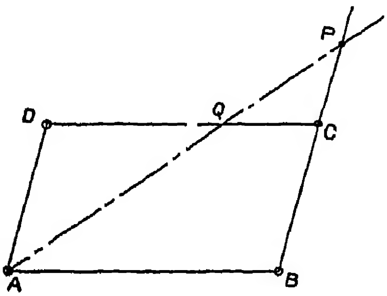


Fig. 75

allows P to approach and recede from A, and the pivot at A permits the whole mechanism to swing round the tracing, point P can reach any point within the limits of the apparatus.

The ratio of reduction is varied by moving P along the arm BC and adjusting the position of Q to correspond, the arms being graduated for this purpose. For convenience of construction, the pivot is frequently placed in DC and the reproducing point, which need not be at the joint A, at a position to correspond. In an actual machine it is necessary that the axial lines of the tracer, reproducer, and bearings be maintained parallel by the construction.

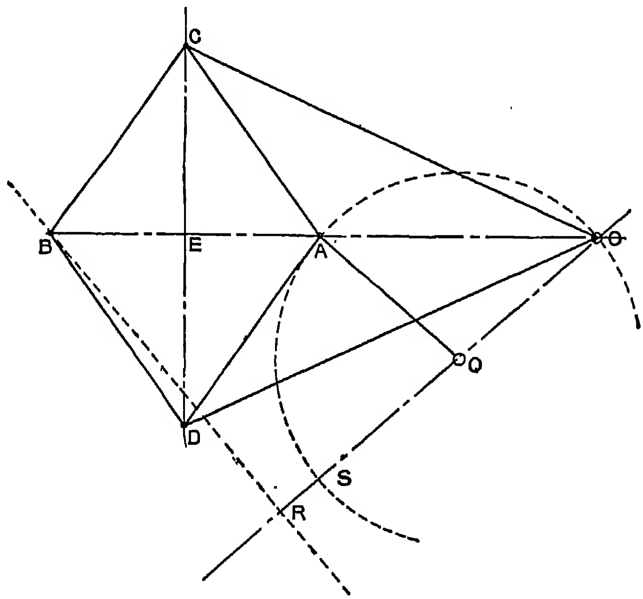


Fig. 76

Straight-line Motions.—In the early days of the steam-engine the production of a straight slide was impossible, and James Watt invented his "straight-line motion" in order to guide the crosshead of his beam engines. In this device the crosshead is connected to a point in a link connecting the end of the beam with a rocking arm by pin joints. The path of this point, over the length of the stroke, is closely approximate to a straight line. Modern facilities for the production of straight planed or bored guides (their accuracy being derived ultimately from the high truth of surface plates) have rendered the device obsolete as regards engines, although it is retained in some instruments where freedom from friction is desired.

Numerous straight-line motions dependent on pin joints only have been devised, but none were mathematically accurate until Peaucellier's cell was invented. In fig. 76 the bars AC, CB, BD, and DA form a jointed parallelogram, the joints C and D being connected by equal links to a fixed

pivot at O. The whole configuration can change its shape, A moving towards O and B away from O, but the joints O, A, B will always lie on a straight line. Since OA.OB $= OE^2 - EA^2$, E being the centre of the parallelogram, the quantity OA.OB is constant since it is equal to $OC^2 - CA^2$, and if A moves in any curve, B moves in the inverse curve. By connecting A by a link to a fixed point at Q where QO = QA, the point A is constrained to move in a circle passing through O. The point B then moves in a straight line BR perpendicular to OQR, since OA.OB = OS.OR owing to the similarity of the triangles OAS and ORB, S being the point where OQR cuts the circle.

Indicator Mechanism.—In engine indicators the pencil point must move in a straight line in such a manner that the amount of its movement

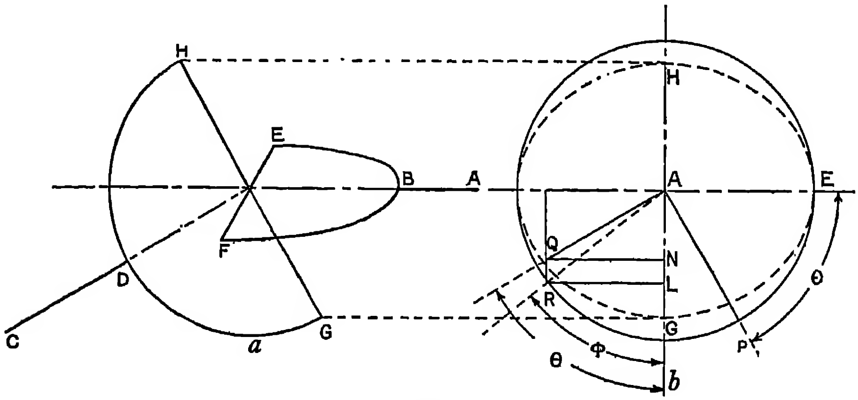


Fig. 77

is proportional to that of the piston of the instrument. Both the pantagraph and Watt's straight-line motion have been employed in the mechanism, but with the increase of engine speed the amount of error due to the inertia of the parts of the indicator have become more serious. The pencil mechanism of a high-speed indicator is shown in Vol. II, fig. 13, p. 91, the moving parts being very light. By taking various equidistant positions of the point A, where the working link is connected to the piston rod, and determining the corresponding positions of the pencil point E, these positions are found to lie equally distant along a straight line to a degree of accuracy quite sufficient for the purpose.

Conic Mechanisms.—In the linkages hitherto considered the axes of the joints have been parallel, i.e. they intersect at infinity. If, however, they intersect at one finite point the links have a corresponding movement; but these mechanisms are infrequently used. Hooke's joint is an important exception.

Hooke's Joint.—The object of the joint is to transmit motion between shafts AB and CD, which intersect at an angle. Each shaft carries a fork (EBF and GDH) with bearings intersecting the shaft at right angles, and a cross having journals at right angles is fitted to work in these bearings

In fig. 77,*a* the axes of AB and CD, inclined at an angle α, and the axis of the fork GH lie in the plane of the paper. In fig. 77,*b* is given a view from the direction AB; the ellipse showing the apparent path of the points GH is the projection of a circle of radius AE, the minor axis thus being AE $\cos\alpha$. Suppose that E moves through an angle θ to P. Then G will appear to move along the ellipse into the position Q when PAQ is a right angle, since the arm G of the cross lies in a plane perpendicular to AP. That is, the point G has moved out of plane GAH a distance QN. The actual angle ϕ through which the shaft CD must turn in moving G to this extent is RAG, where the perpendiculars RL and QN on AG are equal. Since angle QAG is θ, this gives

$$\frac{\tan\phi}{\tan\theta} = \frac{\text{RL}}{\text{AL}}\,\frac{\text{AN}}{\text{QN}} = \frac{\text{AG}}{\text{AE}} = \cos\alpha.$$

The unequal angles turned through by the shafts being connected by this equation, the velocity ratio $\frac{d\phi}{d\theta}$ is given by $\sec^2\phi\,\delta\phi = \sec^2\theta\cos\alpha\,\delta\theta$,

$$\text{or}\quad \frac{d\phi}{d\theta} = \frac{\cos\alpha}{\cos^2\theta\,(1 + \tan^2\phi)} = \frac{\cos\alpha}{1 - \sin^2\alpha\,\sin^2\theta}.$$

The irregularity of this motion, inconvenient in practice, is overcome by the employment of double Hooke's joints, the crosses being properly arranged.

The author desires to thank the following for permission to use various illustrations: Messrs. Longmans, Green, & Co., Ltd. (figs. 4, 10, 25, 26, 27, 34, 35, 63, 64, 65, 66); Messrs. Crosby Lockwood & Son (figs. 5 and 6); Messrs. Macmillan & Co., Ltd. (fig. 24), and the Institution of Mechanical Engineers (fig. 2).

MACHINE DRAWING AND DESIGN

BY

HAROLD DAVIS

Machine Drawing and Design

CHAPTER I

Introduction

Without dwelling too long on the elementary principles of machine drawing, it may be useful to run over the following points, for the sake of those who may have forgotten their early drawing-office training, or of juniors and others whose drawing-office experience is limited.

Instruments, &c.—Intending draughtsmen should equip themselves with the following:—drawing board and tee-square; 60° and 45° set-squares; drawing and tracing instruments; set of divided scales; slide rule, or calculator; protractor and French curves. The first named need rarely be purchased, as they should form part of the office or college equipment.

GOOD DRAWING-OFFICE PRACTICE

Scale Drawing.—It is often observed that some juniors waste a deplorable amount of time in deciding what scale to adopt before commencing a drawing, while others go to the other extreme and do not trouble to think at all, with the result that a drawing requiring fine detail, and which obviously should be drawn to as large a scale as space permits, is drawn so small and with the dimensions so congested that it is almost impossible to read.

If the drawing is in complicated detail, it should be drawn to as large a scale as the standard paper used by the office allows, and, if possible, full size.

Other pleasing scales which meet the requirements of most drawing offices are half-size and 3 in. = 1 ft.

In general-arrangement drawings, such as are required for arranging the position of the machinery and piping in an engine-room, the scale generally used is $\frac{3}{8}$ in. = 1 ft. Other scales sometimes used are $\frac{1}{2}$ in. = 1 ft. and $\frac{1}{8}$ in. = 1 ft.

Dimensions.—There is a tendency of recent years among English and American firms to adopt the decimal system. A great point in favour of this system is that the millimetre, centimetre, decimetre, and metre are multiples by ten of one another. When dimensioning a drawing on the

metric system, all dimensions under one metre are usually given in millimetres. When dimensioning a drawing in feet and inches, a good rule is to express all dimensions less than 2 ft. in inches.

Much time may be saved in the shops by the draughtsman exercising care in dimensioning drawings, and in giving clear indications as to where machining is required. He should also adopt an approved place for the scale, the material to be used, and number required for each detail.

If the stop-valve spindle illustrated in fig. 1 is sent to the shops dimensioned only as at A, the storekeeper must spend valuable time adding these figures together in order to determine the length of the material required. If, however, the total overall dimension is inserted as at B, he can then see at a glance the length required. The above is a very simple example, but if applied to a larger or more complicated detail, such as a rotor shaft, the importance of care will be readily seen.

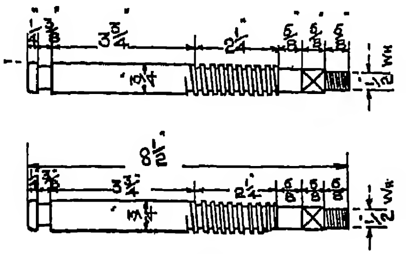


Fig. 1

Machining.—It is important that all working detail drawings of castings and forgings which are sent out from the drawing office to the workshops should clearly indicate where machining is required. Take, for example, the case of a drawing of a casting issued to the pattern-shop, the indications for machining having been omitted. The pattern-maker, when making the wooden pattern, must know where it is necessary to provide for extra metal on the casting for machining. He approaches his foreman, and the foreman in turn must refer to the drawing office for information. This procedure involves a considerable loss of time, a serious consideration now that working hours in the engineering trades are so greatly reduced.

Many firms have different rules for illustrating where allowance for machining is to be left on a casting. The two rules generally accepted are: the etched-line system, where all machined faces are marked thus ⊔⊔⊔⊔⊔; and the preferable and more simple system of marking the machined faces with the letter M or *f*.

The same systems may be applied to sketches sent to the smithy, but with some firms the faces to be machined are indicated by an additional line drawn with a coloured pencil, generally red. When no machining is required, the words " finished black all over " are frequently written on the sketch.

Colouring.—In special cases it is sometimes desirable to colour the finished tracing, different colours representing each metal or alloy. For this purpose best water-colours are recommended, the colours being applied on the back or opposite side of the tracing cloth. It requires some little experience to make this a neat job, and care should be taken to apply the colour in dilute form only, as otherwise a " blobby " effect occurs.

The colouring of drawings is in reality an old custom, when more time

was spent on the appearance and effect of the drawings, but it is rarely done now, except in special cases, such as on new designs submitted to the British Admiralty for approval.

The following list indicates the various colours commonly used to represent different materials:—

Wrought iron	Prussian blue.
Steel	light blue.
Cast iron	Payne's grey.
Brass and gun-metal	gamboge.
Lead	pink.
Wood	burnt sienna.
Stone	yellow ochre.
Brickwork	light red.

Projection.—Frequently views are not correctly projected, and this fault leads to a considerable amount of confusion. Junior draughtsmen

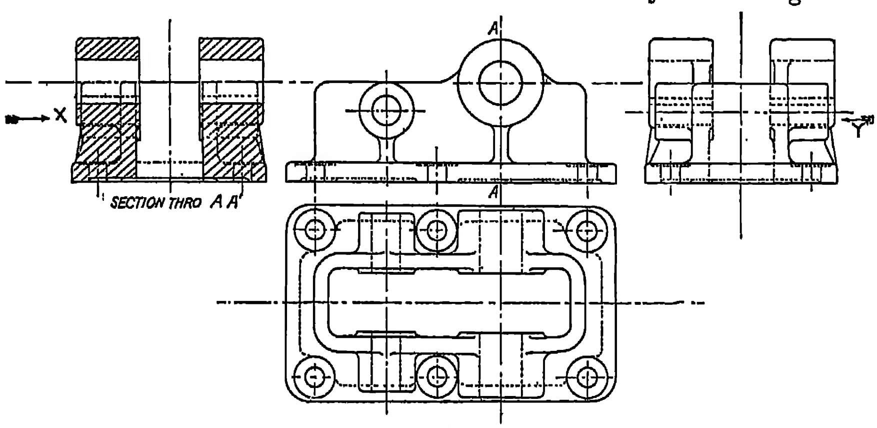


Fig. 2

should always bear in mind that end elevations and sections should be projected and placed at the remote end of the front elevation to which they refer, that is to say, the front elevation or section may be imagined turned over through an angle of 90°. Fig. 2 illustrates this point. The right-hand end view is an end elevation looking in the direction of arrow X, while the left-hand view represents a section looking in the direction of arrow Y. Similarly, in the case of the plan view, the correct position is projected underneath the elevation in question, and *not* above or at the side. Inverted plan views should be used as seldom as possible.

Everybody is aware that dotted lines indicate hidden parts, but many engineers are not aware of the use of the " dot-dash " line. These dot-dash lines show parts non-existent on the view in question, but relate to such parts as would be seen in front of this view. This practice is adopted to reduce the number of views which would otherwise be required in general-arrangement or complicated detail drawings.

Efficient foreman pattern-makers and fitters are cognizant of this pro-

cedure, and are therefore able to read at a glance the true interpretation of the drawing. These lines are distinctive from dimension and centre lines, and should not be confused with same.

Many systems have been introduced for the efficient handling of contracts through the drawing offices and shops. A system, found to be successful to a firm engaged on standardized mass production work, will probably be of little use to a firm engaged on general engineering work. To expedite production, large firms engaged on mass production work make a practice of issuing assembly sketches. These are usually velographs, and illustrate a sectional assembly of the machine. In this case it is only necessary for the draughtsman to add part numbers and a drawing list before issuing to the shops. The feature of this system is that it ensures that no parts are unnecessarily handled in the shops.

CHAPTER II

Bearing Attachments

Generally speaking, bearings may be considered as being of three separate classes, namely, journal, thrust or collar, and pivot bearings.

The word bearing is more often than not undersood to represent the whole attachment, that is, inclusive of the shell or housing and the supporting bracket, whether it be a wall bracket, hanger, or pedestal, but correctly speaking, it should refer only to the bush in which the shaft rotates. All shafts having rotary motion require to be supported in a variety of ways, and to meet these conditions various supports are designed.

A fairly representative number are illustrated diagrammatically in figs. 3 to 10.

A journal bearing may be defined as a bearing on which the thrust or bearing pressure acts in a direction at right angles to the axis of the shaft. Figs. 3 to 8 show examples of such.

In a thrust or collar bearing, the pressure acts in a direction parallel to the axis of the shaft. A bearing supporting a vertical shaft, the shaft terminating at the bearing, is called a pivot or footstep bearing, and here again the pressure acts in a direction parallel to the axis of the shaft.

Proportion of Bearings.—The length of a bearing as compared with the diameter may vary between the ratios of 1·2 to 2·5. This will be governed, however, by the allowable pressure per square inch of projected area, and this varies considerably for different cases in practice. The total load on a bearing is the product of the working pressure and the projected area of bearing surface. The projected area of the journal bearing is the product of the diameter and length, i.e. $d \times l$. The projected area of the thrust bearing illustrated in fig. 11 will be the sum of the bearing areas of the collars.

The essential points to consider when designing a bearing are: (1) whether

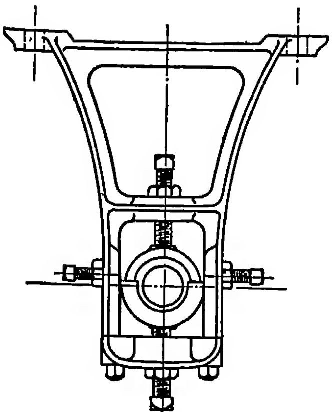


Fig. 3

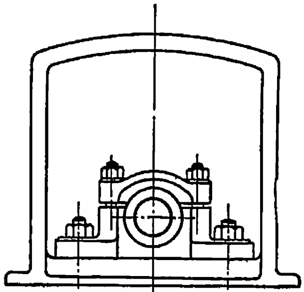


Fig. 4

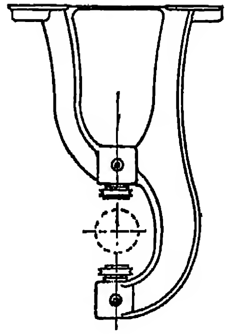


Fig. 6

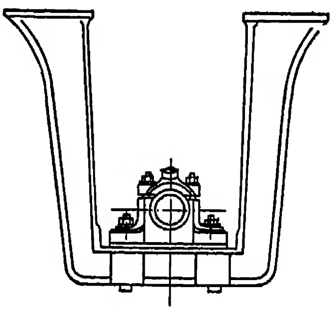


Fig. 5

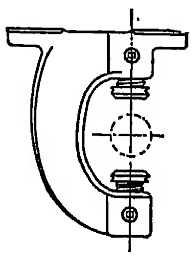


Fig. 7

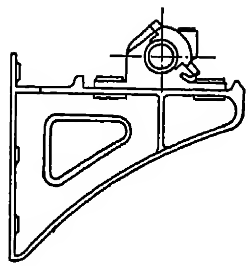


Fig. 8

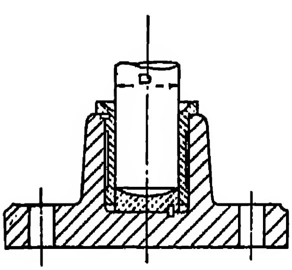


Fig. 9

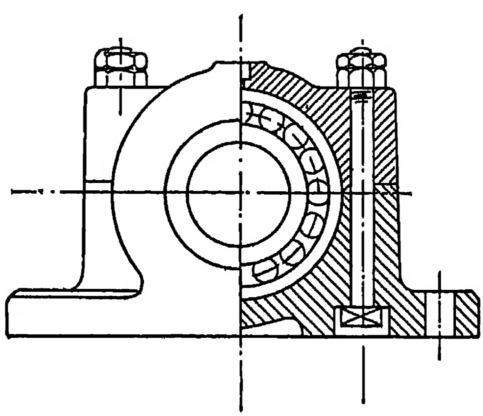


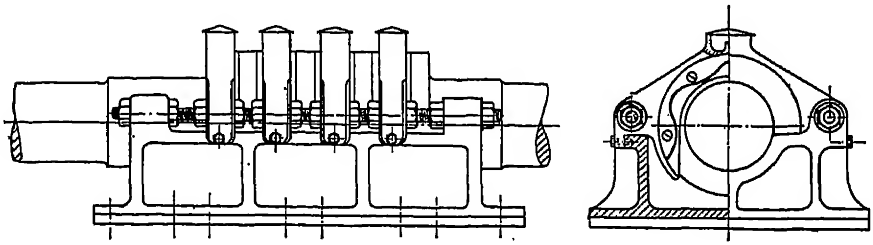
Fig. 10

Figs. 3–10

forced or ring lubrication will be adopted (this will determine the type of pedestal and bearing cap required); (2) the arrangement of suitable oil gauges; (3) facilities for pouring in and draining off lubricating oil; (4) provision of dowels for accurate location in an axial direction. With oil-ring lubrication allow the rings to be 25 per cent immersed. This ensures that an adequate amount of oil will always be carried to the top of the shaft. In the case of oil-ring lubrication, inspection openings in the bearing caps are desirable.

When heavy loads have to be transmitted by vertical shafts the pressure on the pivot may be too high for the bearing to work efficiently, and in such cases a collar bearing is adopted. For this type of bearing, however, the bush must be split, the two parts being fitted to the shaft before lowering it into the pedestal. In a well-designed bearing of this class each collar surface takes a proportional part of the load, and the safe load that may be carried varies with the velocity of the rubbing surfaces.

The thrust bearing illustrated diagrammatically at fig. 11 is of the type commonly used for taking the thrust of the propeller-shaft of a marine



engine. Between each of the collars, horse-shoe shaped pieces of cast steel are fitted; these pieces are lined with anti-friction metal, which allows of periodical renewal to take up the wear. The cast-steel shoes require accurate adjustment, for which provision is made by lock-nuts at both sides of the shoes, the nuts being on two steel screwed bars, one on either side of the thrust block.

Ample provision must be made for lubricating these collars, which work under rather severe conditions, or the anti-friction metal may become overheated and run out. The general practice is to allow the collars to rotate in a bath of oil and soapy water.

The projected area of a pivot or footstep bearing is the area of circle of diameter D, where D = the diameter of the vertical shaft (fig. 9).

$$\text{The total load L} = \frac{p\pi D^2}{4}.$$

The maximum pressure on bearings varies considerably for different cases in actual practice, and whether the load is steady or changing will influence the amount of bearing pressure allowable, as with a changing load the lubricant has more opportunity of reaching the bearing surface.

Working pressures per square inch of projected area met with in practice are as follows:

Main crankshaft bearings	.. ..	300 lb. per square inch.
Crank pins (marine)	.. ..	500 ,, ,, ,, ,,
Crank pins (locomotive)	.. ..	1500 ,, ,, ,, ,,
Gudgeon pins	.. .. ..	800 ,, ,, ,, ,,
Vertical shaft bearings	.. ..	240 ,, ,, ,, ,,
Line shafting ..	.. .. ..	200 ,, ,, ,, ,,

Friction in Journal Bearings.—The safe allowable pressure is dependent upon the temperature of the bearing and the viscosity of the lubricant. With this type of bearing the frictional losses may be calculated, using the following formulæ:

$$\left.\begin{array}{l}\text{Work absorbed in friction}\\ \text{in foot-pounds per minute}\end{array}\right\} = \frac{\mu W \pi D n}{12},$$

$$\begin{aligned} \text{where } W &= \text{total load on bearing in pounds,}\\ D &= \text{diameter of bearing in inches,}\\ \mu &= \text{coefficient of friction (average value ·08),}\\ n &= \text{revolutions per minute.}\end{aligned}$$

$$\left.\begin{array}{l}\text{British thermal heat units}\\ \text{dissipated per minute}\end{array}\right\} = \frac{\mu W \pi D n}{12 J},$$

$$\text{where } J = \text{Joule's equivalent of heat} = 778 \text{ ft.-lb.}$$

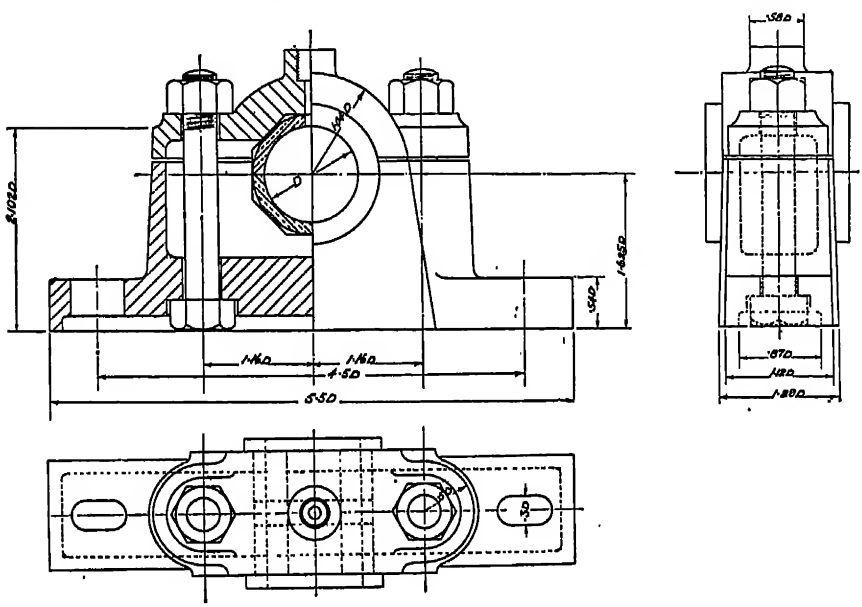


Fig. 12

Plummer Blocks.—A number of firms specialize in the production of plummer blocks, and, generally speaking, the essential dimensions for these

are now standardized. From the standpoint of economy, it is advantageous to use these standard bearings wherever possible, as the manufacturers can produce them cheaply, and can, moreover, deliver from stock. Fig. 12 illustrates a typical plummer block, the dimensions representing the various proportions, the unit being the diameter of the shaft. It will be noted that the brasses are of octagonal form, and are fitted to the cast-iron bracket and keep. On reference to the figure, it will be seen that the casting is lightened out to a marked extent, thus bringing about a considerable reduction in weight with a consequent reduced cost of production.

Hanger Bearings, Wall-brackets, &c.—Figs. 3, 5, 6, and 7 illustrate various standard types of hanger bearing, the name hanger being given to all bearing arrangements which support lengths of shafting from a ceiling or an overhead beam. The majority of hanger bearings are adjustable, thus simplifying the setting or "lining up" of the shaft or countershaft.

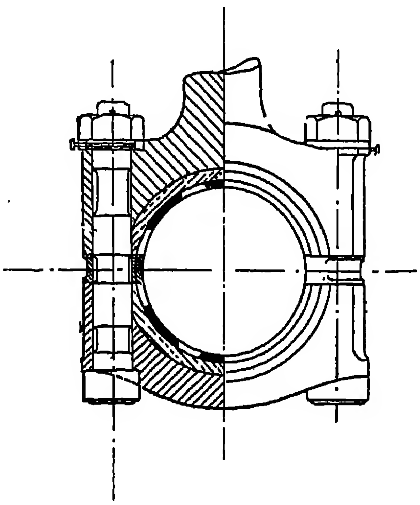


Fig. 13

When arranging shafting in a building, the draughtsman, having first decided the position of the shafting, makes provision for its support. If the shafting is in close proximity to the floor, it may be supported by ordinary plummer blocks secured thereto, that is, if the pulleys are of small diameter and there is sufficient clearance. Usually, however, the shafting is supported from the wall by wall-brackets with a minimum overhang; and to these wall-brackets the plummer blocks are secured.

A wall-bracket with plummer block attached is shown in fig. 8.

A shaft revolving in a plane parallel to the surface of the wall may be supported by a wall-box (see fig. 4). An opening is cut in the wall to accommodate the wall-box with its plummer block. This arrangement may also be used to support a line of shafting extending to an adjacent room.

Brasses.—For ordinary purposes bearings are fitted with inner bushes in halves (commonly called " brasses ", though actually made of gun-metal or phosphor-bronze). These may vary considerably in design.

For good-class work either the bearing shells or the brasses are lined with anti-friction metal, the inner surfaces of the shells being dove-tailed to hold the metal more firmly in place (see fig. 13). This alloy is soft, and may be scraped and fitted to the specific running clearance desired. It is important, however, that the inner surface of the shells should be tinned before the white metal is run in, otherwise bad contact will be made between the inner surface of the shell and the metal.

Overheating in a bearing may often be traced to a bad contact between the anti-friction metal and shell. The alignment of bearings, and particularly of main bearings, is of great importance, as the running performance and, in fact, the life of the engine depend on this factor.

Main bearings should be relined from time to time, otherwise many serious results, a broken crank-shaft, for example, may occur.

In the case of an engine fitted with a fly-wheel, the bearing adjacent thereto will usually require remetalling sooner than the others.

The following procedure should be adopted when remetalling bearings. Having first melted out the old white metal, boil the bearing shells in caustic soda to remove oil and grease, then rub over with a wire brush. The shells must then be heated and tinned, after which the white metal may be poured in evenly. The once-common practice of hammering is detrimental to the metal, as it only tends to loosen the same.

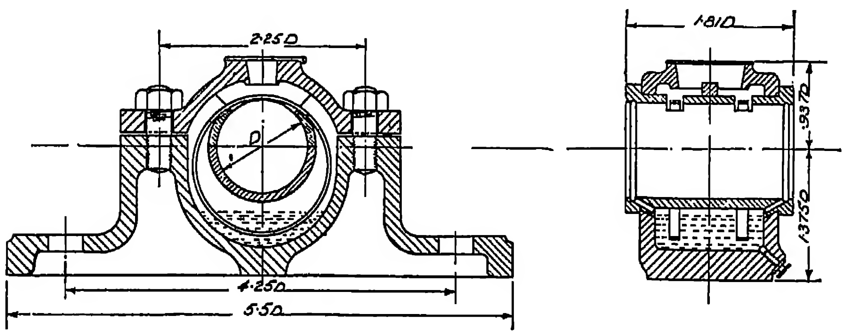


Fig. 14

Hot bearings may result from using lubricating oil of inferior quality, or even be due to the oil-channels in the bearings being cut at the wrong spot, thereby destroying valuable bearing surface. Care should be taken to filter all lubricating oil to keep it free from dust and grit.

The bearings of a new engine, or an engine the bearings of which have been relined, require close attention for the first few hundred hours' running, after which they should give little trouble, unless, of course, they are neglected.

Miscellaneous Bearings.—The bearing illustrated in fig. 14 is an improvement on the ordinary pedestal bearing, being fitted with a ring lubrication device. The rings revolve, and during rotation dip into an oil-bath formed in the bearing casting; the oil is thus constantly fed on to the top of the shaft and to the bearing.

Two annular passages, one at either end of the brasses, allow the oil to return to the bath, thus eliminating waste. Some manufacturers provide these bearings with a gauge-glass at the side, so that the depth of oil in the bath can be verified at a glance.

CHAPTER III

Shafts and Rotating Parts

The following are points to be considered when determining the diameter of a shaft. Firstly, when the shaft is subjected to a uniform twisting moment only, known as torque. By equating the torque to the moment of resistance to twisting, the diameter of shaft may be evaluated.

$$T = \frac{\pi d^3}{16} f_s.$$

$$\text{Then } d = \sqrt[3]{\frac{T \times 16}{\pi f_s}},$$

where T = torque in inch pounds,

f_s = maximum shear stress (in pounds per square inch),

d = diameter of shaft in inches.

The torque may readily be found from the known horse-power absorbed and the revolutions per minute at which the shaft rotates.

$$\text{Torque} = \frac{\text{horse-power} \times 33000 \times 12}{2\pi N}.$$

Secondly, considering the shaft to be subjected to combined twisting and bending, when for shafts of circular section the diameter may be determined from

$$T_1 = M + \sqrt{(M^2 + T^2)},$$

where T_1 = equivalent twisting moment,

M = bending moment.

The maximum bending moment is calculated in the usual manner by treating the shaft as a beam. The loads are usually due to the weight of pulleys or wheels, or is the thrust on a crank or cranks according to their number. Equating T_1 to the moment of resistance to twisting, d, the diameter of the shaft, may be found.

The case of a shaft subjected to bending only presents little difficulty. The bending moment is calculated in the usual manner, and, by equating this to the moment of resistance to bending, the diameter of shaft d is found.

$$M = \frac{d^3\pi}{32} f, \text{ or } d = \sqrt[3]{\frac{10{\cdot}2M}{f}}.$$

Hollow shafts are extensively used when shafts of large diameter are required, as a reduction of approximately 20 per cent may be gained in

weight without reducing the strength of the shaft. The torque in a hollow circular shaft may be calculated from the following formula:

$$T = \frac{\pi}{16}\frac{(D^4 - d^4)}{D}f_s.$$

Stiffness.—It is important to remember that stiffness is proportional to the fourth power of the diameter of the shaft, and inversely proportional to the length. For ordinary mill-shafting an angle of twist of 0·1° per foot of length may be allowed.

Shaft Supports.—The maximum distance in feet between bearings for a shaft carrying an average number of pulleys is $= \sqrt{32d}$, where d = diameter of shaft. For shafts subjected to no bending action, other than that due to its own weight, this distance may be increased by approximately 50 per cent.

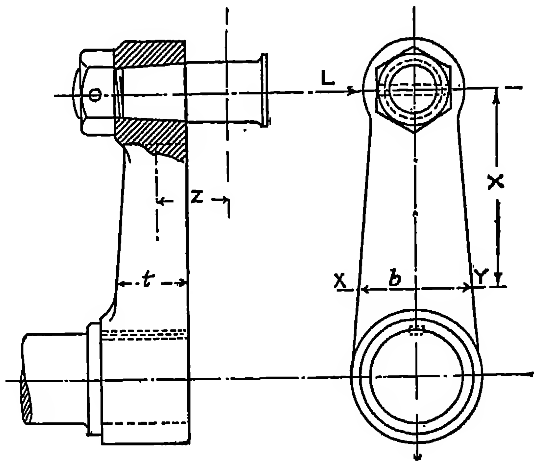


Fig. 15

Cranks and Crank-shafts.—Among the various types of cranks in use some are forged solid, while others are built up owing to the difficulties experienced when forging. For small petrol-engines and for heavy oil-engines the crank-shaft is solid forged, but built up in large marine steam-engines. This has the advantage that, should one portion of the shaft fail, a spare length may be fitted without scrapping the entire shaft.

The following points are to be observed when designing a small overhung crank as in fig. 15. The usual practice is to shrink the crank on to the journal of the shaft; the crank-pin may either be riveted over at its tapered end or secured by a nut and split-pin.

Considering the bending actions acting on any section, firstly, where bending is parallel to the plane of rotation, then

$$Lx = \frac{b^2t}{6}f,$$

$$\text{and } t = \frac{6Lx}{b^2f},$$

where L = maximum force acting on crank-pin with crank and connecting-rod at right angles,

f = stress in pounds per square inch,

x = distance from crank-pin to section,

z = distance from centre of crank-pin to web centre.

Secondly, when crank is on the dead centre, then

$$L_1 x = \frac{bt^2}{6} f,$$

$$\text{and } t = \sqrt{\frac{6L_1 x}{bf}}$$

when L_1 = force on crank pin in dead-centre position.

A single- and a 4-crank type of solid shaft are shown in figs. 17 and 16.

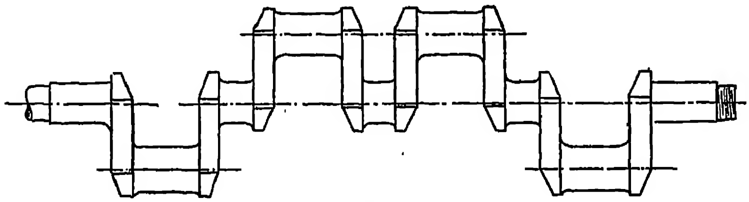


Fig. 16

A drawing of a solid forged crank-shaft, suitable for a Diesel oil-engine, is shown in fig. 18. It will be noticed that holes are drilled through the webs from the crank-pin journal to the main bearing journal to provide for a forced lubrication system. The crank-angles are at 120°, and are arranged for a six-cylinder engine, and as in this case the balance is good, balance weights are not required. For two-cylinder engines balance weights are generally provided to compensate for the inertia forces of the reciprocating and rotating masses, or there

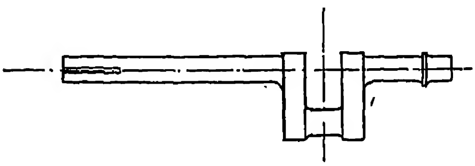


Fig. 17

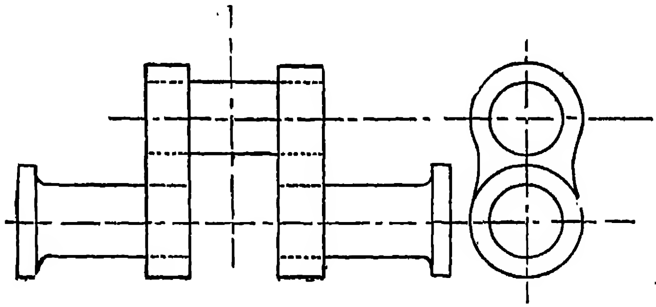
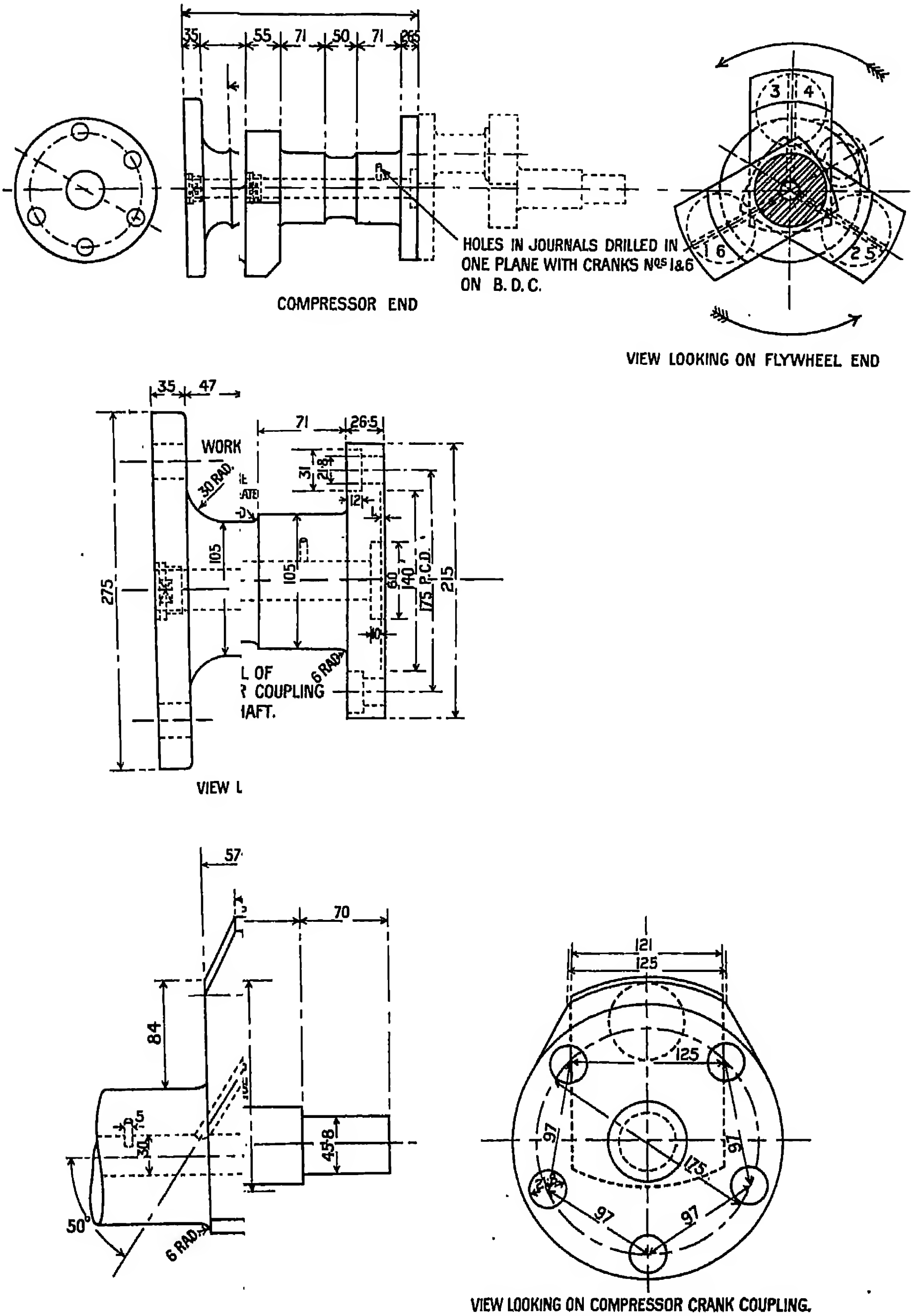


Fig. 19

would be some danger of vibration affecting the engine foundations, with subsequent trouble with the alignment of the main bearings.

Crank-shafts for Diesel engines require periodical realignment to forestall fracture, and it is customary for insurance companies, when inspecting their clients' engines, to take readings relating to the wear-down of the


HOLES IN JOURNALS DRILLED IN
ONE PLANE WITH CRANKS Nos 1&6
ON B.D.C.
COMPRESSOR END
VIEW LOOKING ON FLYWHEEL END
WORK
30 RAD.
6 RAD.
COUPLING
P.C.D.
6 RAD.
VIEW LOOKING ON COMPRESSOR CRANK COUPLING.

crank-shaft bearings two or three times a year. Two methods of checking the alignment of such shafts are as follows: one by gauging between the webs with the crank at top and bottom centres, and the other by measuring with a micrometer the thickness of the main bearing shells between the journal and the bed-plate housing.

Built-up Cranks.—A form of built-up shaft is shown in fig. 19, such as is extensively used for marine engines. Another type is shown in fig. 20.

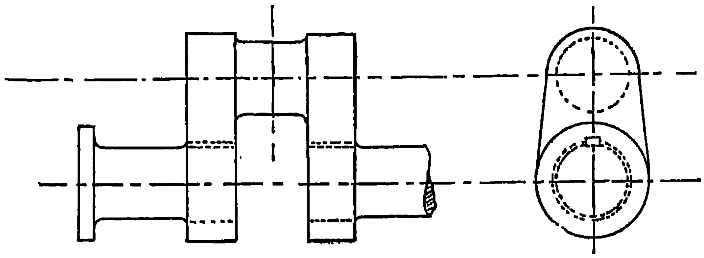


Fig. 20

Here the crank-pin end webs are in one piece, and may be either keyed to the shaft or bolted to same. The diameter of marine-engine shafts may be calculated from the following formulæ issued by the Board of Trade:—

$$S = \sqrt[3]{\frac{C \times P \times D^2}{f\left(2 + \frac{D^2}{d^2}\right)}}, \text{ for compound condensing engines,}$$

$$\text{and } S = \sqrt[3]{\frac{C \times P \times D^2}{3f}} \text{ for ordinary condensing engines,}$$

where S = diameter of shaft in inches,
C = length of crank in inches,
D = diameter of low-pressure cylinder in inches,
P = absolute pressure in pounds per square inch,
d = diameter of high-pressure cylinder in inches,
f = constant (see following table).

Angle between cranks (2 cranks) ..	90°	100°	110°	120°	130°	140°	150°	160°	170°	180°
Crank and propeller shafting, f = ..	1047	966	904	855	817	788	766	751	743	740
Tunnel shafting, f = ..	1221	1128	1055	997	953	919	894	877	867	864

For one crank take f for 180°. For three cranks at 120° take f = 1110 for crank and propeller shafts and 1295 for tunnel shafting.

Shaft Couplings.—For connecting two lengths of shafting of small diameter, the box- or muff-type couplings, as shown in figs. 21 and 22, are

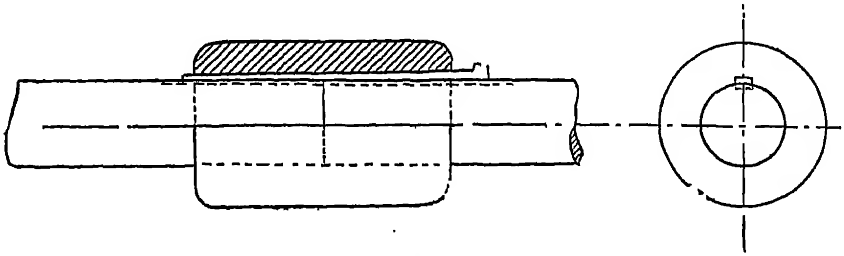


Fig. 21

generally used. It will be seen that a cast-iron sleeve slides over the two ends of the shafts which butt together.

The split-muff coupling (fig. 23) has found favour, due to the fact that

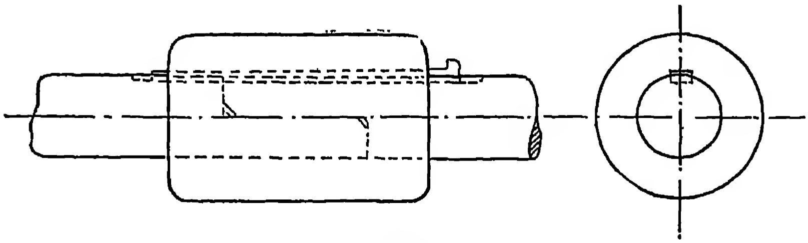


Fig. 22

it can be quickly fitted or detached. The box is in halves and held together by four or six bolts, and tightened up to clip the shaft. The coupling is prevented from turning on the shaft by a sunk key. For temporary work

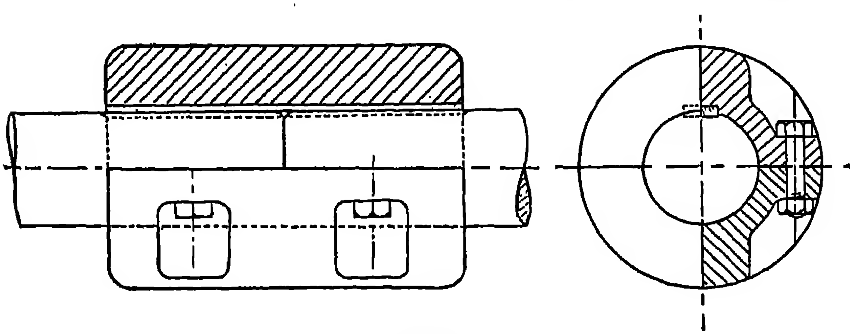


Fig. 23

wood couplings without keys are fitted. Such couplings may be used as pulleys, as both the heads and nuts of the bolts are recessed.

For better-class work a flange coupling is adopted (see figs. 24 and 25); this type of coupling is made in either cast iron or steel, and is keyed to the end of the shaft. To ensure true alignment it is essential that the joint

made by two flanges should be accurately machined, and, if possible, should be of spigot-and-socket form. The coupling bolts should be a good driving fit, and in the case of large shafting it is usual to drill the holes in couplings

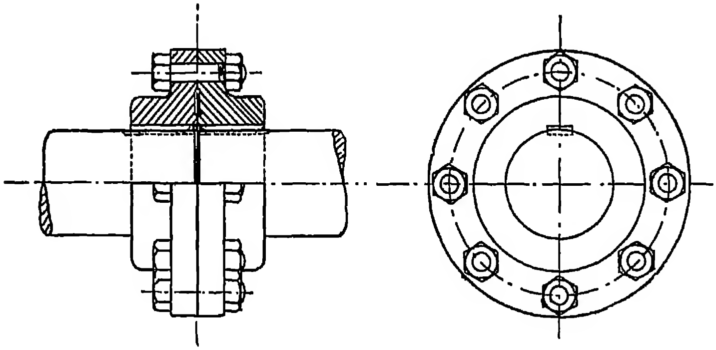


Fig. 24

below finished size and then reamer out after lining up. For marine work the flanges are sometimes forged solid with the shaft.

Strength of Coupling Bolts.—It is usual when calculating the diameter of coupling bolts to equate the moment of resistance to shearing

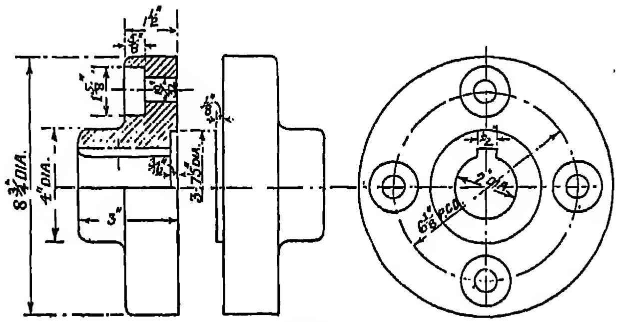


Fig. 25

of the bolts to the moment of resistance or torque of the shaft. Allowing a stress of $\frac{3}{4} f_s$ in the bolts, as there may be partial bending, we have

$$\tfrac{3}{4} f_s \frac{\pi d^2}{4} \mathrm{NR} = \frac{\pi \mathrm{D}^3}{16} f_s.$$

$$\therefore d = 0{\cdot}577 \sqrt{\frac{\mathrm{D}^3}{\mathrm{NR}}},$$

where D = diameter of shaft in inches,
d = diameter of bolt,
f_s = shear stress in pounds per square inch,
N = number of bolts,
R = radius of pitch-circle.

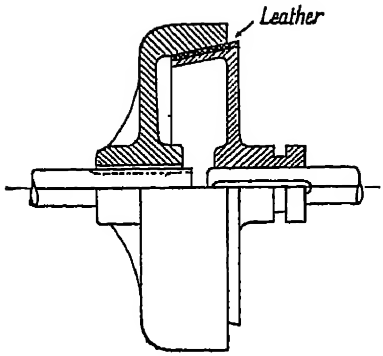


Fig. 26

A form of coupling or friction clutch as adopted for motor-car work is illustrated in fig. 26. The inner cone or male portion is lined with leather, and may be renewed from time to time. To engage or disengage the clutch the operator releases or holds down a pedal operating a system of levers, thus giving the male or cone portion a lateral movement. Multiple-disc clutches of the Hele-Shaw type are, however, largely used for motor-car work.

CHAPTER IV

Connecting-rods and Bolts

The connecting-rod, as its name implies, is the link between the reciprocating motion of the piston and the rotary motion of the crank-shaft. To speak of the length of the connecting-rod is to refer to the distance measured from the centre of the crank-pin to the centre of the crosshead or gudgeon-pin. The ratio of this length of the connecting-rod to the length of the crank radius varies between 2·5 and 5, but this limit may be sometimes exceeded in an exceptional case. Connecting-rods are usually made of forged steel, and may be either circular, rectangular, or of H section.

Strength of Connecting-rods.—Designers and draughtsmen should have ample data available when determining the diameter of a connecting-rod, based on the experience dearly bought by early engineers. In the case of circular rods the diameter may be calculated from Molesworth's formula, viz.:

$$d = 0{\cdot}021\text{D}\sqrt{p} \text{ for iron and } d = 0{\cdot}018\sqrt{p} \text{ for steel,}$$

where D = diameter of cylinder in inches,
d = diameter of rod in inches,
p = pressure in cylinder, pounds per square inch.

The thrust of the connecting-rod for any position of the crank is P secθ, where P is equivalent to the total load on the piston in pounds and θ the angle made by the connecting-rod with the line of stroke. In cases where it is desirable to investigate the stresses set up in the rod it is usual to apply one of the formulæ for long columns deduced by Euler, Rankine, Gordon, and other authorities. The Rankine-Gordon formula may be expressed as follows:

$$P = \frac{fA}{1 + 4\frac{l^2}{d^2}k},$$

where P is the ultimate load,

f, the stress in pounds per square inch,

A, the cross-sectional area in square inches,

k, a constant,

l, the length of rod in inches,

d, the diameter of rod in inches.

In this formula both ends of the rod may be considered as hinged; then k is $\frac{1}{3000}$.

Rods of circular section are common in marine work, but in the case of long or high-speed connecting-rods, they are usually made to one of the sections shown in fig. 27.

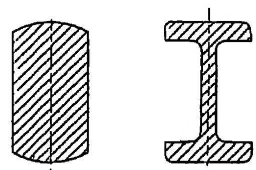


Fig. 27

In high-speed engines the bending action introduced by the inertia of the connecting-rod should always be carefully considered and due allowance made to counteract. The flexural stresses, arising from the oscillations of the rod in the plane of revolution of the crank, vary with the speed of the engine and the position of the rod, and may often exceed the tensile and compressive stresses due to the load on the piston. When such is the case, the connecting-rod is best adapted to resist strain when designed in the form of a tapered beam of I or rectangular section having its greatest depth at the crank-pin end. Locomotive connecting- and coupling-rods are typical examples of correct design in this respect. The same remarks apply to the connecting-rods of aero-engines, where it is of special importance to secure the maximum of strength with the minimum of weight.

Connecting-rods are usually tapered, in the case of vertical engines, from the small-end to the big-end bearing, this gradual change of section being essential for formation of the forged ends, and to give strength to the centre portion of the rod where the bending stress is greatest.

Fig. 28 represents a connecting-rod suitable for a marine engine. One end, it will be noted, is forked with its bearings fitting one on either side of the crosshead, while the other end is connected by bolts, termed big-end bolts, to the big-end bearing and crank-shaft. The big-end bearing brasses are lightened out and have distance-pieces fitted between them, these distance-pieces having a slot cut in them to clear the bolts. As wear takes place the brasses may be remetalled, and accurate adjustment made by the insertion of thin liners of sheet metal. Lubrication of the big-end bearing is effected by feeding the oil-box on the top end of the rod, the oil being transmitted by gravity to the bearing through a small pipe connected by clips to the side of the rod.

In certain locomotives, the connecting-rod is long with a resultant small angular motion, and it is found satisfactory to take up the longitudinal adjustment of the brasses by the well-known wedge-and-cotter arrangement. There are various attachments of the crank-pin end in use, as, for example,

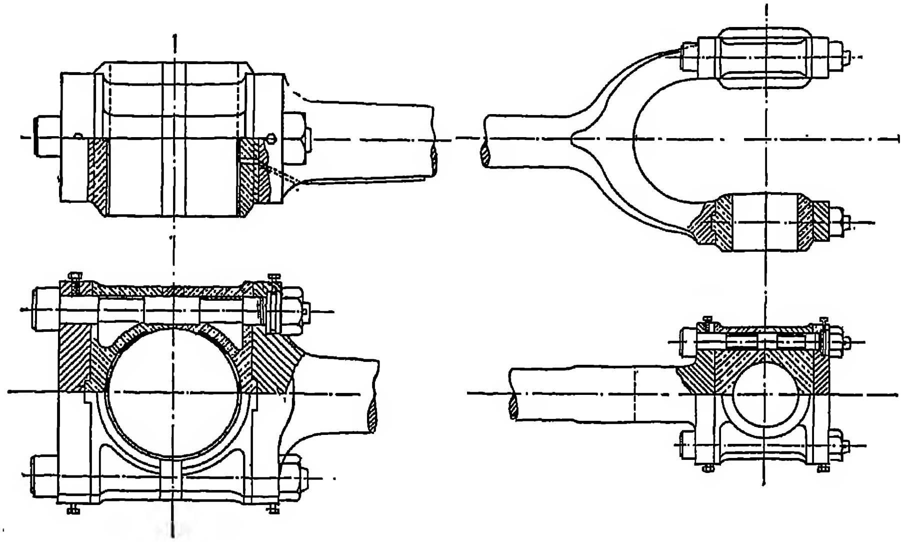


Fig. 28

the box-end and the strap type. The crosshead end of the rod is frequently made solid, similar to the top end of an oil- or petrol-engine. Examples of the crank-pin and crosshead ends of a connecting-rod are shown in figs. 29 and 30.

Two typical connecting-rods, representative of the type usually adopted

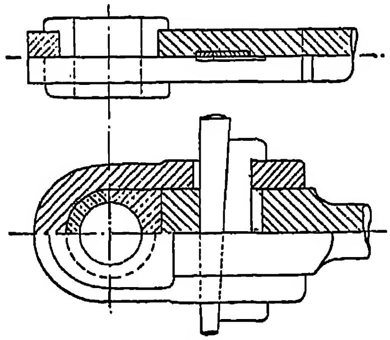


Fig. 29

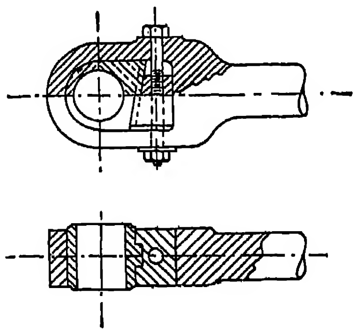


Fig. 30

for oil- and petrol-engines, are illustrated in figs. 31 and 32 respectively. It will be observed that the former is hollow, which admits of transmitting lubricating oil under pressure to the top-end bearing. It is sometimes necessary in oil-engine work to alter the length of the connecting-rod slightly

for adjustment of the compression volume, and with a rod of the form shown in fig. 31 this may readily be done by inserting thin sheet-metal liners between the foot of the rod and the big-end brasses. While touching on connecting-rods for heavy oil-engines, it may not be amiss to remind designers that in a vertical type engine the connecting-rod is generally drawn out with the piston through the cylinder, after first removing the cylinder cover and valve operating gear. To make this possible the overall dimensions of the big end must be sufficiently small to pass through the cylinder.

The connecting-rod shown in fig. 32 is of a type used for automobile work,

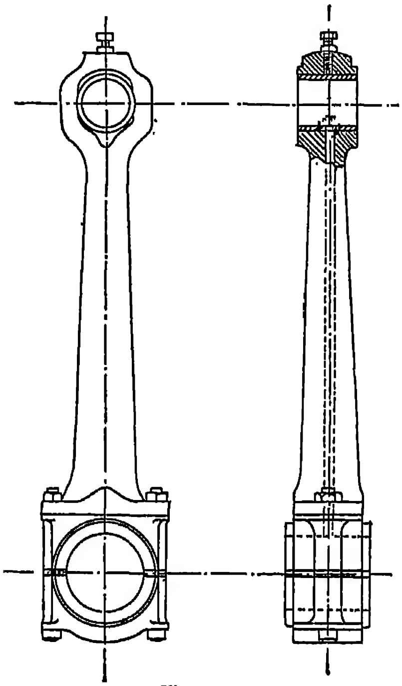


Fig. 31

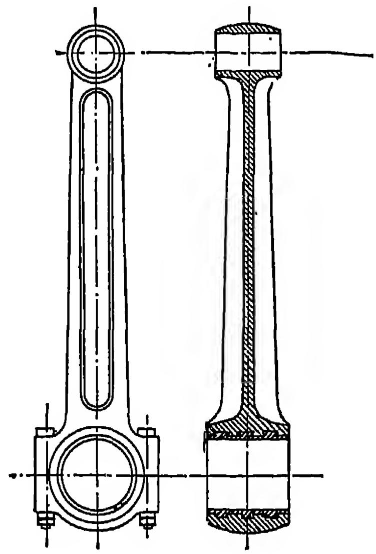


Fig. 32

and is usually of the familiar beam section as shown. It has the advantage of lightness without loss of strength, and can be readily produced as a stamping. The big end is lined with anti-friction metal, while the small end is not bushed. This latter is first drilled to a diameter of about fifteen thousandths of an inch smaller than the finished bore, and finally broached or reamered to size.

The connecting-rods of aero-engines vary considerably in design on account of the several types of engines in vogue, but can be classified as two main types, i.e. rotary and stationary engines. The connecting-rods for the stationary type engine are of beam or H section, and are made of chrome-nickel steel, with the big ends lined with white metal and the small ends with phosphor-bronze bushes. The distinctive feature of the connecting-rod for rotary aero-engines is, that in place of the conventional big-end bearing a forged-steel attachment called a " shoe " is fitted. With this

type of engine the several connecting-rods are attached to discs having grooves in which the shoe ends of the rods oscillate. In this case the virtual big end is common to all the rods, and rotates on two radial ball-bearings mounted on the stationary crank-pin (see fig. 33).

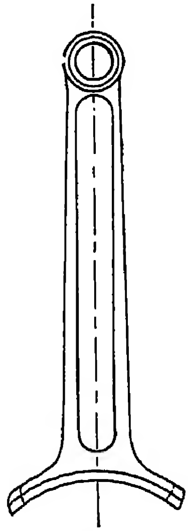


Fig. 33

Another form of attachment applied to rotary engines is that with one master connecting-rod, to which the other auxiliary rods are attached by pins, technically termed wrist-pins, this master-rod running on two bearings.

Connecting-rod Bolts.—It is usual to design connecting-rod bolts and the crosshead bolts sufficiently strong to carry the maximum thrust on the rod, and, having in view that these bolts work under very severe conditions, being repeatedly in tension and compression, it is essential for them to be made amply strong and of the very best material to withstand the consequent shock. It is known that breakdowns have frequently occurred through failure of these bolts, particularly in heavy oil-engine work, hence it is desirable that they be made of steel 3 per cent nickel, and, wherever possible, should follow the design as shown in fig. 34. The obvious feature of this is that the diameter at the bottom of the thread is no weaker than the reduced diameter; also this practice makes it that the fitting portion or collars on the bolt may be more quickly machined than a bolt of parallel diameter throughout.

Several papers have been read of late on the failure of connecting-rod bolts and the presumed beneficial results gained by heat treatment. After a certain number of working hours the steel in the bolts may become fatigued and should be renewed, the safe number of working hours, of course, being dependent on the quality of the steel used. Some engineers advocate periodical annealing, but unless this is done by the actual steel-makers with good metallurgic data as to the composition of the steel, good results cannot be guaranteed.

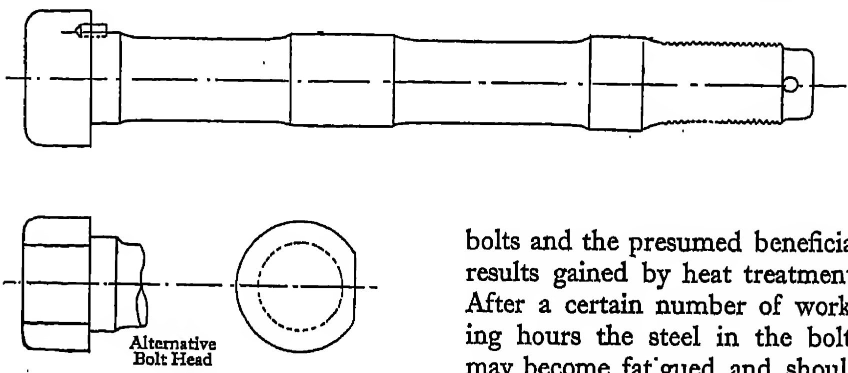


Fig. 34

CHAPTER V

Pistons and Piston-rods

Pistons are an essential part of a reciprocating engine, whether its working medium be steam, gas, or oil. If the piston be of a diameter equal to that of the piston-rod, it is termed a plunger. Pistons are usually made of cast iron or cast steel, but in special cases forged steel is used. When cast iron is used, the best close-grained metal only should be considered, and the casting should be free from blow-holes, and in the case of gas- and oil-engines care should be taken to ensure sound metal at the point of concentration of temperature and pressure, that is, obviously, at the piston crown.

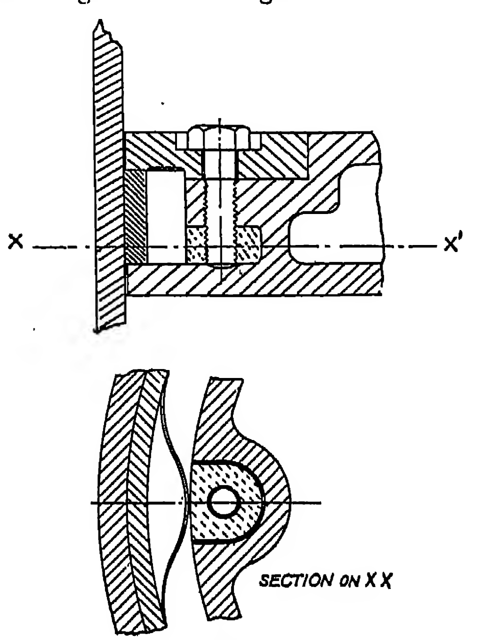


Fig. 35

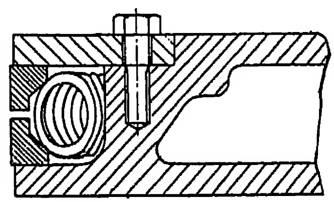


Fig. 36

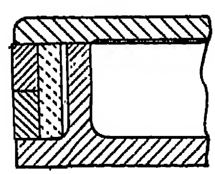


Fig. 37

Pistons vary considerably in design, so only the types most common in general practice will be considered. All pistons, excluding plungers, are fitted with packing rings of varied arrangement, but always to a common purpose, viz. to prevent any leakage of the working substance past the piston. Such rings are of rectangular section, are split, and are fitted into grooves turned in the main body of the piston. The rings are sprung into the grooves, so that when the piston is placed in the cylinder, the rings press outwards against cylinder walls, thus making a practically tight joint to prevent the working substance from blowing past. As a further safeguard against a blow past, the joints of the rings are arranged at different circumferential positions on the main body or piston-skirt.

For pistons of large diameter, such as are found in steam-engine practice,

the split ring previously described would be scarcely suitable, as the piston must necessarily be withdrawn when at any time it is required to renew a ring. This difficulty is overcome by fitting rings known as junk-rings, several types of which are shown in figs. 35, 36, and 37.

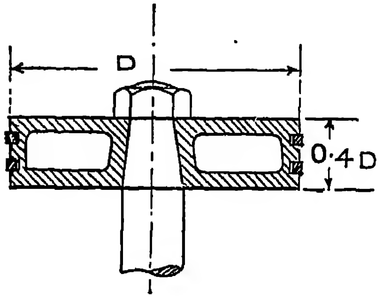


Fig. 38

In one type illustrated it will be observed that dished springs are placed at intervals behind the piston-ring, so as to exert a pressure in a radial direction tending to press the split ring more tightly against the cylinder wall. There are many such devices in use; sometimes a helical spring or a piece of corrugated steel is substituted for the dish springs described.

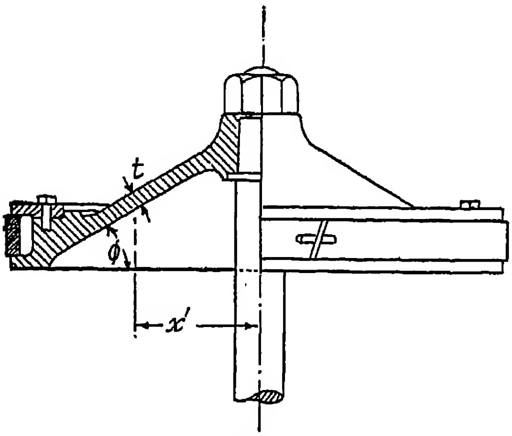


Fig. 39

These piston-rings are held in place by the junk-ring secured to the main piston casting by set-bolts. Great care should be taken that these bolts are securely tightened down, and that the steel used in the bolts is of the very best quality.

Figs. 38, 39, and 40 illustrate usual types of piston as employed in steam-engine practice. It will be noted that the depth is comparatively shallow as compared with the diameter, and is made hollow in section and reinforced by ribs. With the piston conical in form, a reduction may be made in the prevailing thickness of metal.

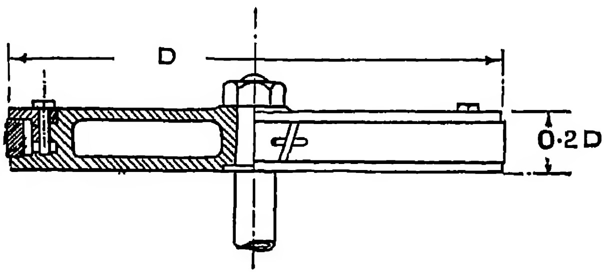


Fig. 40

Thickness of Conical Pistons.—Mr. J. Kraft has investigated the strength of conical pistons (*Proc. Inst. Civil Engineers*).

Let p = pressure of steam in pounds per square inch,

f = allowable stress pounds per square inch (3000 for cast iron, 9000 for steel),

T = minimum thickness of metal at a point near the boss, x inches from centre.

t = minimum thickness of metal at any other point x' inches from centre (see fig. 39),

R = radius of piston.

$$\text{Then } T = \frac{p}{2f \sin \phi} \times \frac{R^2 - x^2}{x}, \text{ and } t = \frac{px'}{f \sin \phi}.$$

Where the dishing is very small use the formula for flat plates.

Pistons for Internal-combustion Engines.—It is common to design pistons sufficiently long to act as a crosshead guide, except in the case of large engines where a crosshead is fitted. The connecting-rod is secured direct to the piston by means of a case-hardened pin known as a gudgeon- or wrist-pin. The usual length of such trunk pistons is generally between the limits 1·25 to 2·5 D (where D = diameter of piston).

On reference to fig. 41, showing a section of a piston suitable for a 4-cycle Diesel oil-engine, it will be seen that the piston crown is very thick. This is necessary on account of the severity of the conditions under which it works. In modern practice the crown is dished as shown, and frequently a core-plug device, which can be renewed from time to time, is fitted to the centre. The gudgeon-pin is stepped as shown in fig. 42.

These pins are secured by set-bolts through the bosses provided on the interior of piston. It is essential that the gudgeon-pin should be adequately lubricated, as trouble here may lead to piston seizure.

It is recommended that at the small-end bearing, the bearing pressure should not exceed 800 lb. per square inch of projected area.

In practice the piston is tapered up from the bottom obturator ring, while the lower part of the piston, which acts as the crosshead, is machined parallel.

Expansion of the metal in the vicinity of the gudgeon-pin may cause a piston seizure, or a slight variation from the truly circular form may occur when driving the gudgeon-pin into place. As a precaution against this the external surface of the piston about the gudgeon-pin holes is relieved, or the inside bore of the cylinder is barrelled out to about a maximum of 0·020 in.

Large pistons are generally made with detachable heads, so as to ensure good castings and to cut down expense should renewal be necessary.

The proportions given in figs. 41–44 are a fair average for Diesel gas- and petrol-engines, though in the case shown in fig. 41 the thickness of the piston crown is usually taken as approximately 12 per cent of the cylinder diameter.

Piston-ring Joints.—The joints usually adopted for rings of small

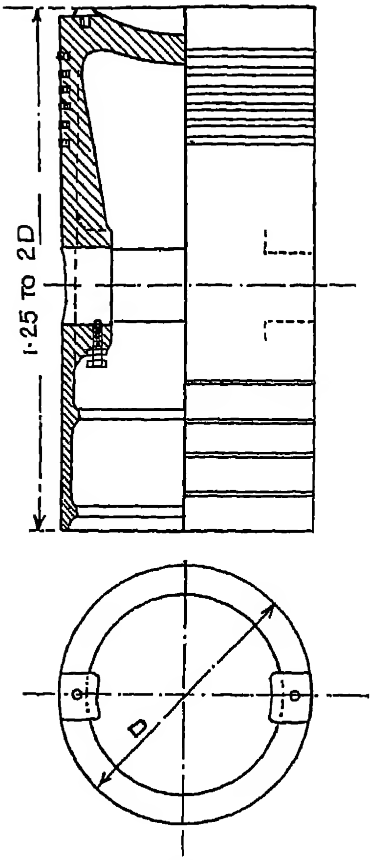


Fig. 41

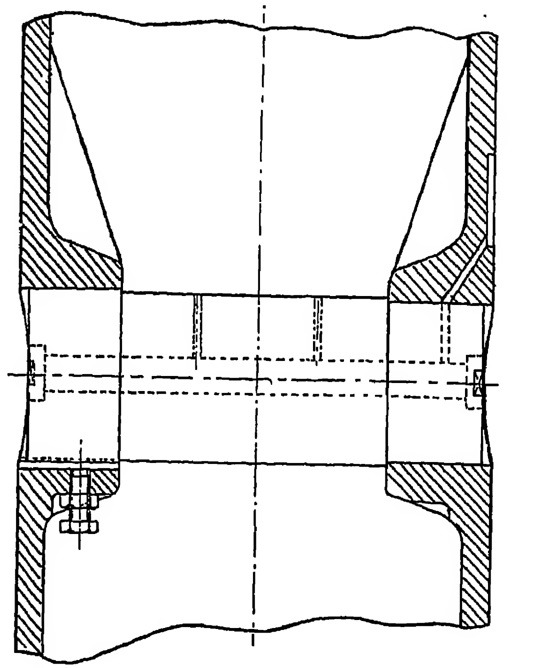


Fig. 42

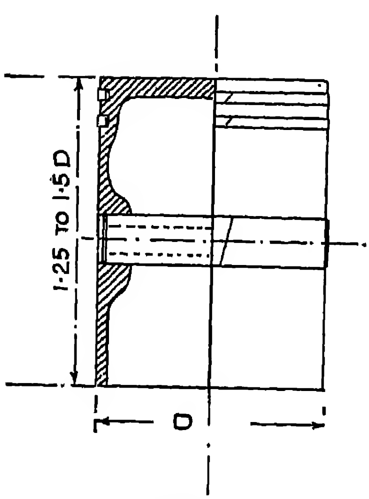


Fig. 44

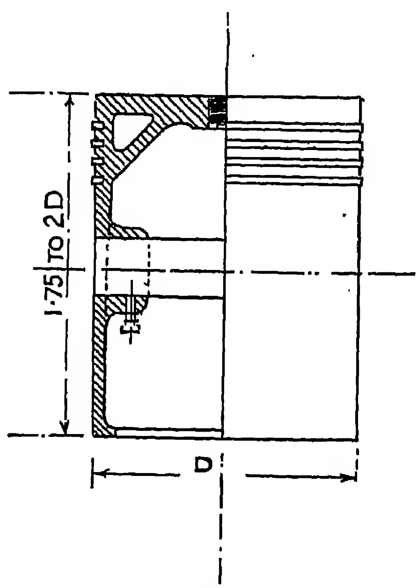


Fig. 43

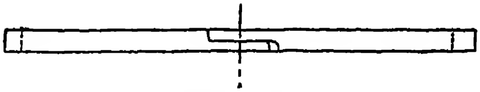


Fig. 45.—Lap

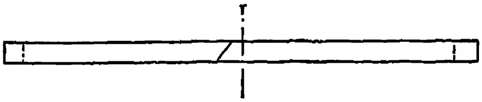


Fig. 46.—Bevel

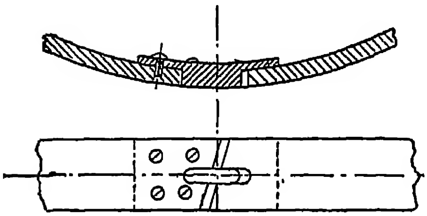


Fig. 47

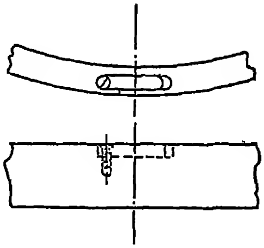


Fig. 48

Piston-ring Joints

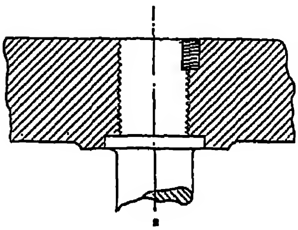


Fig. 49

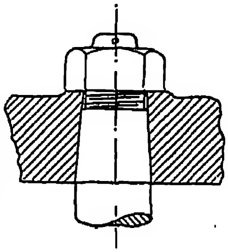


Fig. 50

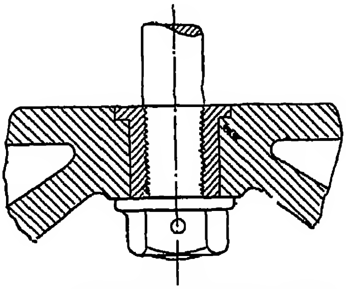


Fig. 51.—Conecting Upper and Lower Piston

Connection of Piston-rod to Piston

diameter are the bevel cut and lap, as illustrated at figs. 45 and 46, but in the case of large rings a tongue piece or similar device is fitted to prevent the working substance from blowing past. Two devices frequently adopted for steam-engine work are shown in figs. 47 and 48. Some engineers prefer rings of very small diameter to be made eccentric, the reason for this being to prevent breakage when springing the ring into its groove. Small rings are first turned slightly larger in diameter than the cylinder bore and then cut, but with a large ring of over 18 in. diameter the external diameter is left rough turned, then, after the gap is cut out, the ring is closed to the working gap clearance, and the outer diameter finally turned to suit the cylinder bore. Such rings are usually made of cast iron, and the iron must be homogeneous, hard, and close grained to ensure long life.

The patent hammered cast-iron rings manufactured by The Standard Piston Ring Co. are very satisfactory, as it is found that automatic hammering

on the inner periphery of the ring produces the uniform tension which ensures perfect fitting, and also a longer life than with the ordinary type of ring.

Piston-rods.—Various methods of securing the piston-rod to piston are shown in figs. 49–51. Small pistons may have the rod screwed in, and a grub screw fitted to prevent it from slacking back. A more satisfactory job is obtained with the nut attachment (see fig. 50).

As piston-rods are subjected alternately to tension and compression, it is customary in ordinary practice to allow a stress not exceeding 4500 lb. per square inch.

Molesworth gives the following formula, from which d, the diameter of the rod, may be calculated:

$$d = \mathrm{CD}\sqrt{p},$$

where d = diameter of rod in inches,
D = diameter of cylinder in inches,
p = pressure in cylinder, pounds per square inch.

C is a constant whose value varies with the ratio

$$\frac{\text{length of rod}}{\text{diameter of rod}} = \frac{l}{d}.$$

The following table gives values of C for various values of $\frac{l}{d}$.

$\frac{l}{d}$	10	15	20	25	30
C	·0164	·0171	·0181	·0193	·0207

CHAPTER VI

Crossheads, Guides, and Eccentrics

The name given to that part of an engine intermediate to the piston and the connecting-rod is the crosshead. There are three types of crosshead used respectively in marine, land, and locomotive work, but it must be understood that in practice this classification is not rigid.

A type of crosshead as used for marine work is illustrated in fig. 52. It will be seen that the crosshead is of the slipper type, and consists essentially of a gudgeon-pin shrunk into the crosshead block or forged solid with same. To this is attached both the piston-rod and the guide shoes, the latter being generally made of cast iron or cast steel with anti-friction metal strips dovetailed into them.

Every sea-going engineer learns, at an early stage of his experience, the importance of keeping the guides and shoes well lubricated. Failure

to do this will result in an overheated guide, with the danger of the anti-friction metal melting out. It should be noted that with this type of crosshead the forked end of connecting-rod is secured to the gudgeon-pin journals by brasses arranged for convenient adjustment.

A type of crosshead popularly used for stationary or land engines is shown in fig. 53. The external portion of the crosshead is accurately turned to suit a guide of cylindrical form, which construction obviously cheapens the cost of production in comparison with that requiring flat surface guides. The slippers are forged solid with the crosshead, the whole attachment being secured by a cotter to the piston rod.

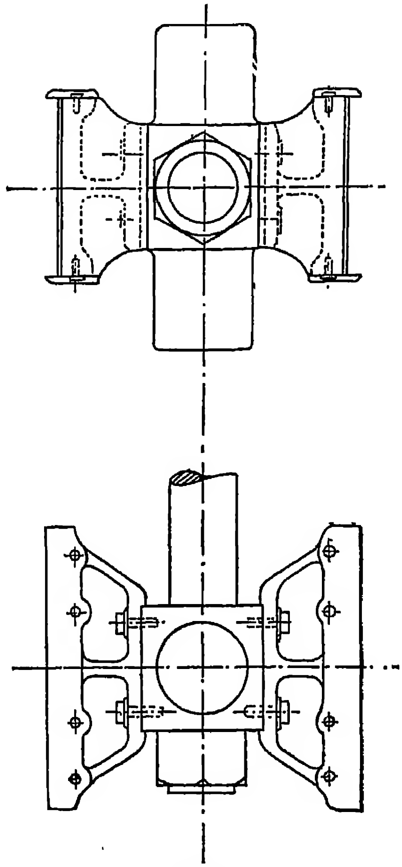


Fig. 52

A type of crosshead frequently adopted in locomotive practice, designed by Mr. Adams of the G.E.R., is shown in fig. 54. A distinguishing feature from the previous examples is, that the whole attachment embraces a guide bar of rectangular section, the cover plate being attached with the bar in place. To minimize friction these rubbing surfaces are also lined with anti-friction metal, and, as will be seen in figure, provision is made for adequate lubrication to the slide bar and pin. Another type of crosshead adopted in locomotive practice is illustrated in fig. 55.

The bearing pressure at the crosshead guide, due to obliquity of con-

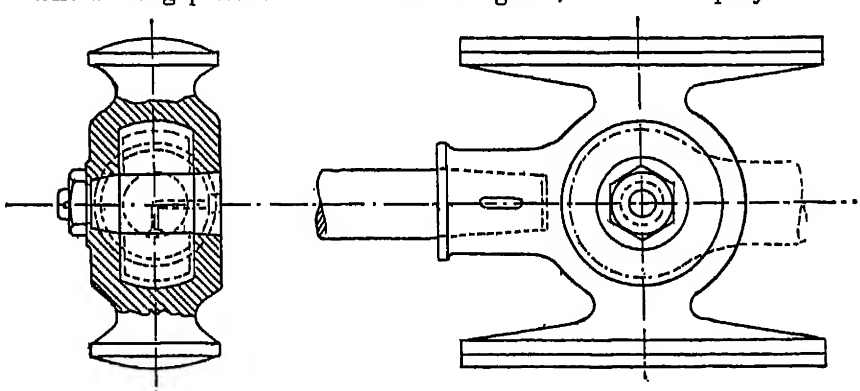


Fig. 53

necting-rod, should be from 50 to 80 lb. per square inch for marine work, and 40 to 50 lb. per square inch for stationary or land engines.

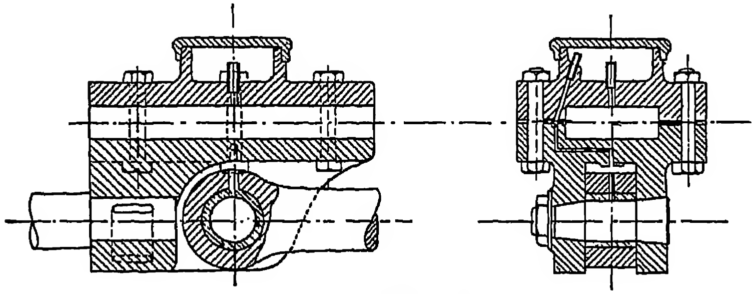


Fig. 54.—Adams's Crosshead

Diameter of Gudgeon-pin.—Consideration will now be given to the proportions of the pin shown in the marine-type crosshead (see fig. 52), where the journals are at each side. In the chapter on connecting-rods

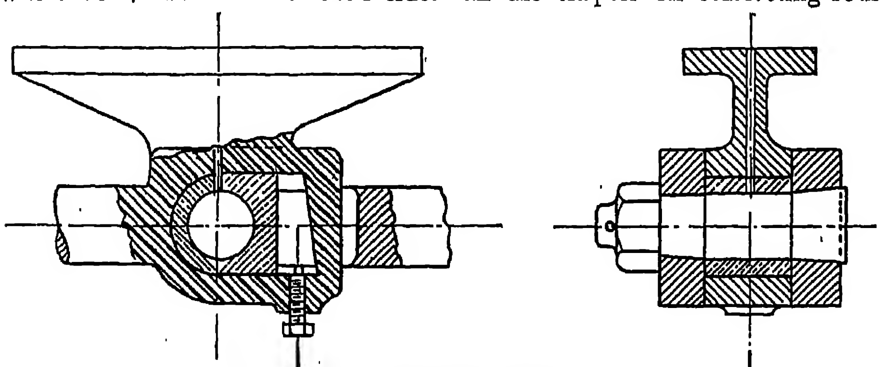


Fig. 55.—Stroudley's Crosshead

(p. 74) it is stated that the thrust is equal to P $\sec\theta$. If this load is assumed to be divided equally between the two journals, then

$\frac{Tl}{8}$ = bending moment, so that by equating the bending to the moment of resistance we have

$$\frac{Tl}{8} = Zf, \text{ and}$$

$$d = \sqrt[3]{\frac{1{\cdot}273\,Tl}{f}},$$

where $Z = \frac{d^3\pi}{32}$,

and f = stress in pounds per square inch,

l = total length of bearing journals.

Eccentrics.—The function of the eccentric is to convert the rotary motion of the crank-shaft to reciprocating motion for the slide-valve spindle. The eccentric consists of a sheave and strap, both in halves. Sheaves are

generally of cast iron, the halves secured by cotters or bolts, the whole being embraced by the strap, which in turn makes connection with the eccentric rod foot. The sheave is prevented from turning on the crank-shaft by means of a key, while as a further safeguard against longitudinal movement set screws are fitted. To obviate any possible bending tendency the crown of the strap is sometimes made of greater thickness than the lower half.

Proportion of Sheaves.—The breadth of a sheave is determined from the total thrust necessary to overcome the resistance of the slide valve, and

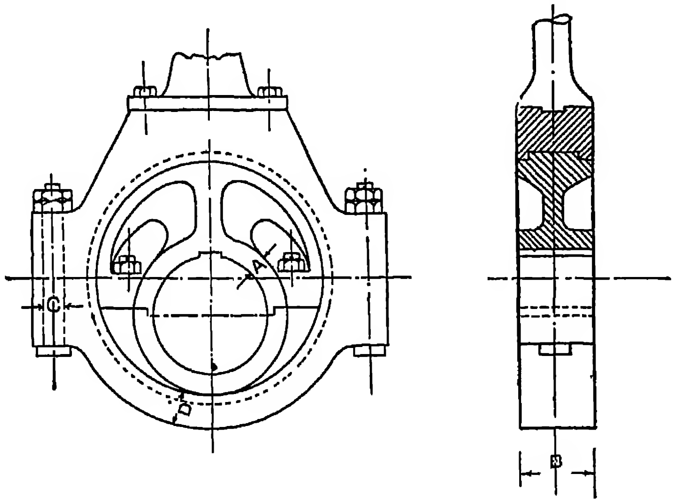


Fig. 56

it is usual to allow a bearing pressure of about 60 lb. per square inch for land engines, and 70 to 140 lb. per square inch for marine engines. As a guide in designing eccentrics the following proportions in terms of the breadth may be found useful. (See fig. 56.)

Thickness of metal A = 0·56 B.
Diameter of bolt C = 0·4 B.
Thickness of metal D = 0·5 B.

CHAPTER VII

Cylinder Covers

For convenience in manufacture, and cheapness in production, covers for steam-engine work are made in cast iron and secured to the main cylinder casting by steel studs or bolts.

The cover casting is well ribbed to give the necessary strength with

lightness, and is frequently made of conical form, thereby allowing the thickness of metal to be reduced without a consequent weakening of the cover.

When designing a cover, the two essential computations to be determined are: the prevailing thickness of metal, and the diameter and number of bolts or studs required to form a steam- or gas-tight joint under the working pressure in operation. Considering first the thickness of a flat cast-iron cover, assuming the diameter of cylinder bore and flange to be fixed. This may be found by the following formula for circular flat plates, given by Unwin:

$$f = kp\left(\frac{r}{t}\right)^2,$$

where p = working pressure, pounds per square inch,
f = working stress, pounds per square inch,
r = radius, in inches,
k = 0·5 for mild steel,
k = 0·8 to 1·2 for cast iron,
t = thickness of metal in inches.

In the case of a dished or conical cover the thickness may be calculated from the formula of Mr. J. Kraft, as given on p. 81, this formula being applicable to either dished covers or pistons. A good rule for approximate results is to make the thickness of cover = 1·25 times the thickness of the cylinder wall. The number of studs or bolts required for a cover joint may be determined by equating the total load on the cover to the sum of the total strengths of the studs.

Let d = diameter of bolt at bottom of thread in inches,
D = cylinder diameter in inches,
p = working pressure in pounds per square inch,
f = stress in pounds per square inch,
N = number of studs.

$$\text{Thus } \frac{\pi D^2 p}{4} = \frac{\pi d^2 N f}{4},$$

$$\therefore d = D\sqrt{\frac{p}{Nf}}.$$

In high-pressure steam work a good general rule is to make the distance between each stud equal to 3·5 times the stud diameter, and for very low-pressure work equal to 5 times the diameter of bolt or stud.

When calculating the diameter and number of bolts, the following safe allowable stresses may be applied for steel studs or bolts.

Ordinary practice f = 5000 lb. per square inch
Small cylinders f = 2500 lb. " "

When the joint is often broken it is usual to allow a stress of only 2000 lb. per square inch, for it will be appreciated that the bolts or studs will be subjected to severe loads owing to overstressing when tightening up. A typical cylinder cover, suitable for the L.P. cylinder of a marine engine, is illustrated in fig. 57. In steam-engine practice cylinder covers are not necessarily always circular and of the form shown in figs. 58 and 60, but, for example, may be formed to accommodate a part of the steam port of the cylinder (see fig. 59).

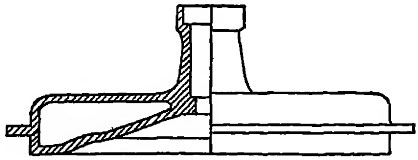


Fig. 57

Cover or Head for Heavy-oil Engine (4-cycle type).—The cover, or cylinder head as it is frequently called, of a heavy-oil engine is an extremely important and interesting

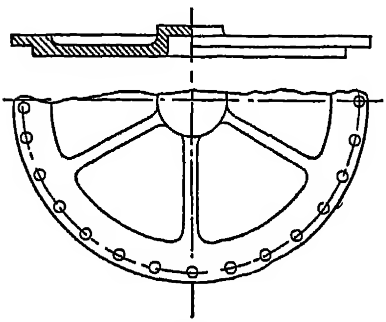


Fig. 58

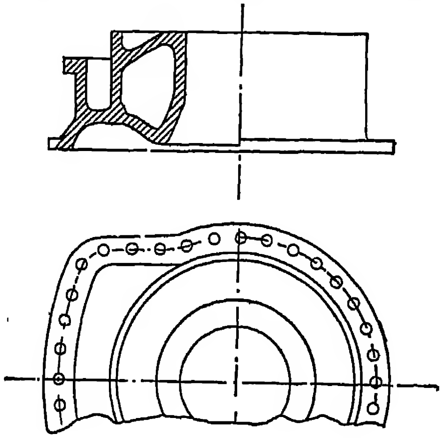


Fig. 59

part of the engine, and, as it contains pockets and chambers forming the seatings for the fuel, air, and exhaust valves, it will be seen that an elaborate casting is required. These covers are usually of cast iron, though sometimes cast steel has been employed in spite of the practical difficulties in procuring sound castings of this metal; manufacturers, however, generally prefer to revert to cast iron.

By reference to fig. 61 (where two views of a typical cover suitable for a 4-cycle Diesel engine are shown) it will be noted that the cover is of cylindrical form, flat on the top and bottom surfaces. On the bottom surface is the spigot which makes the joint between the cover and cylinder liner, the whole cover being securely fastened down by studs. The half-sectional plan shown indicates the intricacies of the air and exhaust passages and the seatings of fuel and air starting valves, also the several inspection doors, through which the sediment deposited by the circulating water may be cleaned out. It may

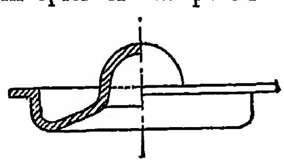


Fig. 60

be mentioned here that these covers are invariably water-cooled, and in the latest modern practice the circulating water, under pressure of about 8 to 12 lb. per square inch, is led by specially designed internal pipes and fittings direct to the vicinity of the hot zone, which is in close proximity to the centre of the cover, where it impinges on the metal about that point.

The pockets for the various valves must leave a rather weak section at

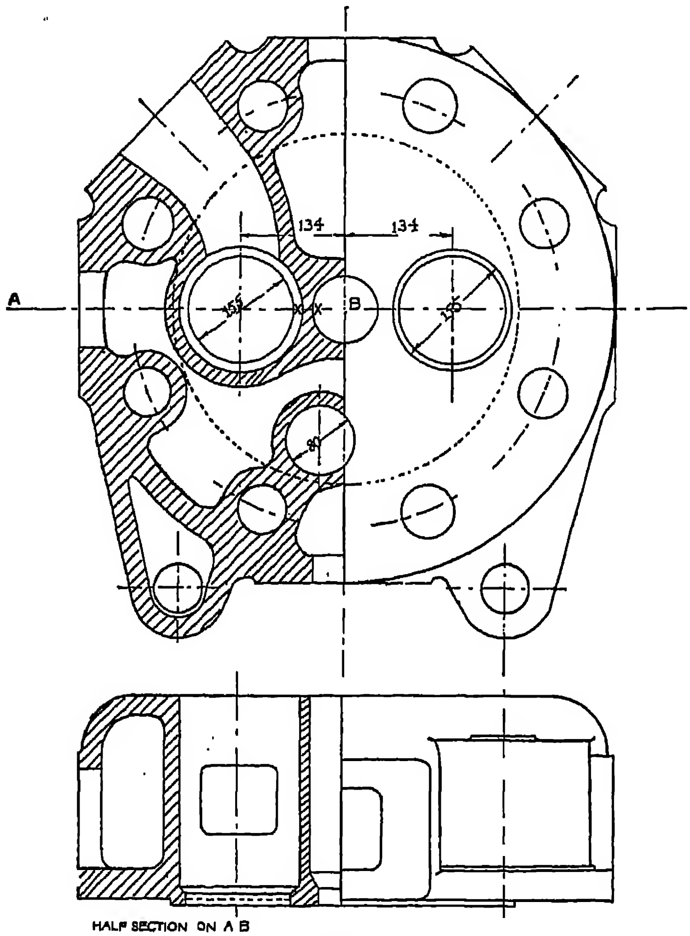


Fig. 61

X X (see fig. 61), and it is about this point covers usually fail. Some manufacturers place the centre pocket (i.e. for the fuel valve) out of centre; this undoubtedly is a possible advantage tending to lengthen the life of the cover.

For marine work a cylinder cover of box section is preferred, and in this case long bolts are used to secure the cover to the cylinder body. These bolts are made long in order to reduce the stresses, and are arranged to pass through tubes expanded into the cooling-water space of the cover.

The passages and parts of a cover suitable for a 2-cycle engine differ

somewhat from the 4-cycle cover illustrated in fig. 61. If the engine is designed for valve scavenging, the cover contains, in addition to the fuel injection and air starting valves, two or four air scavenging valves.

The thicknesses of the various walls, ribs, &c., of a cylinder cover are based on the valuable practical experience gained with pioneer engines, and on account of the irregular sections, both in cylindrical and cross section, combined with the continuously variable expansion and contraction stresses set up therein, it is somewhat difficult to arrive at reliable figures for calculation. We may, however, check very approximately the stresses in portions of the walls by calculating the maximum bending moments and the moduli of the sections, both for cylindrical and cross sections. The lower plate is always made thicker than the other portions of the cover, being as it is in direct contact with hot gases and subject to great pressure, while it must support the several valve-case seatings which open into the combustion space, and possibly extra small holes drilled to accommodate indicator cock, relief valves, &c.

CHAPTER VIII

Valves

Much ingenuity has been displayed by engineers in designing the many and varied types of valves encountered to deal with their sundry working conditions, but the definite function of any valve is to establish communication at approved intervals between one receptacle and another.

Broadly speaking they may be classified under three heads, namely, slide valves, lift valves, and hinged valves, but each of the three types may

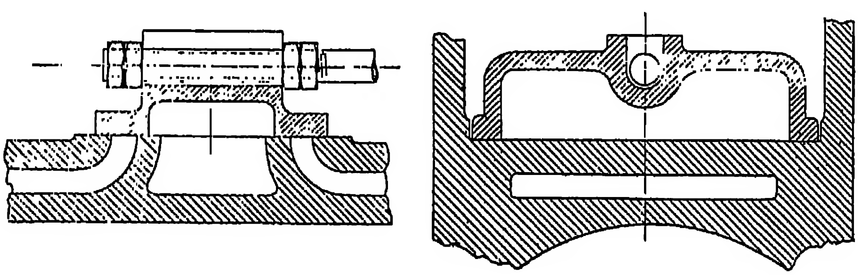


Fig. 62

be again subdivided almost indefinitely. Of course it is understood that the word valve in the foregoing classification is used in a general sense, referring collectively to the valve, its seating, and any chamber or casting that may exist for the purpose of carrying the seating and guiding the valve, or its operating gear.

Flat and Piston Slide Valves.—In a steam-engine cylinder both the flat (single or double ported) and the piston slide valve are extensively

used for distributing the steam. The flat slide valve is of box form, and is operated by an eccentric on the crank-shaft which gives it a reciprocating

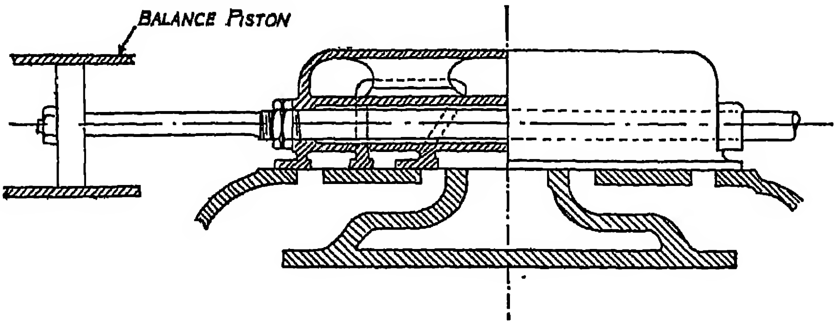


Fig. 63

motion, thereby covering and uncovering rectangular ports, thus regulating the supply and cut-off of steam to the cylinder. In actual practice the valve overlaps the edge of the steam port when in the middle position of its travel. The amount by which the outer edge of the valve overlaps the edge of the steam port is called "outside lap", while where the inner edge of the valve overlaps the steam port, this amount is termed "inside lap". The result of introducing this lap is, in the case of the former, to cut off the steam after a desired proportion of the piston stroke, and to rely on the expansive properties of the steam to complete the stroke, and in the case of the inside lap to delay the release of the exhaust steam so that the compression is augmented at the end of stroke. By the aid of the Bilgram or the Zeuner diagram the outer and inner laps may be readily determined, also the valve travel, and the relative position of the valve for any position of the crank. Figs. 62 and 63 indicate a single- and a double-ported valve respectively.

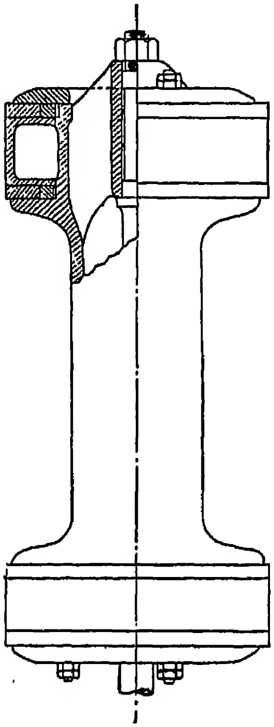


Fig. 64

Piston slide valves have found favour due to the fact that they give greater economy owing to the frictional resistance being considerably less than with the ordinary flat valve, while the cylindrical form may be more accurately machined and fitted. The action of the valve is similar in every respect to the ordinary flat valve, the reciprocating motion covering and uncovering ports cut in the walls of the cylinder, any leakage or "blow past" being prevented by the employment of piston rings as on the ordinary piston. By reference to fig. 64 it will be seen

hat the trunk or main body is a casting, generally of cast iron, the centre ortion being hollow to permit free passage of the steam; the valve rod s attached to the centre boss as shown. The distance pieces and piston ings are kept in place by flat circular plates, which are secured to the main body by studs.

Lift Valves.—Everyone is familiar with the common lift valve, such

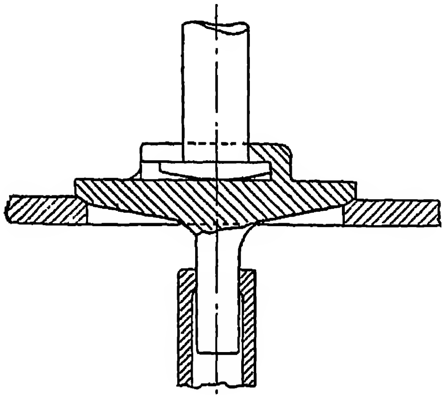


Fig. 65

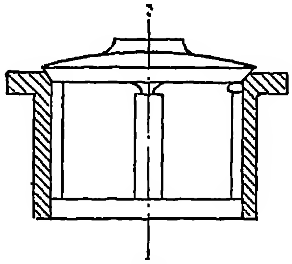


Fig. 66

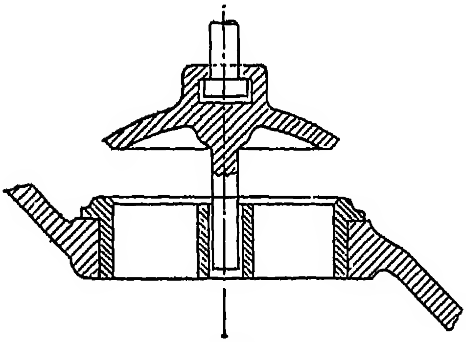


Fig. 67

as is adapted in various forms to control steam, gas, oil, water, air, &c. The valve itself is of circular form with ribs or legs cast or machined thereon, the function of which is to guide the valve when returning to its seat.

The circular part of the valve is machined to a mitre, generally of 45°, on which a joint is made with the corresponding mitre on the valve seating. When assembling the valve and accompanying parts this joint is ensured by carefully grinding in the valve with a little carborundum or emery powder. This should be done from time to time as wear or "pitting" takes place, otherwise the valve will cease to be effective. The general method of operating these valves is by means of a spindle screwed through a nut made in the valve-chamber cover or bridge piece, so that by turning a hand-wheel, or similar attachment fitted to this spindle, the valve may be opened, closed, or regulated.

Figs. 65–68 represent various types of valves and seatings as used in ordinary steam practice. In selecting a valve the buyer would be well advised to choose a type in which both the valve and its seating may be renewed without dismantling either the main valve body or the adjacent pipes and fittings. In this type the seating is sometimes screwed into the main valve body, and may be extracted by a special spanner. Fig. 68

illustrates a well-designed renewable-type valve manufactured by Messrs. Dewrance & Co., Ltd. A stop valve having a spindle screwed into the bridge piece external to the body is preferable, as the thread is free from contact with the working substance, and, with reasonable care, may be kept clean and well lubricated.

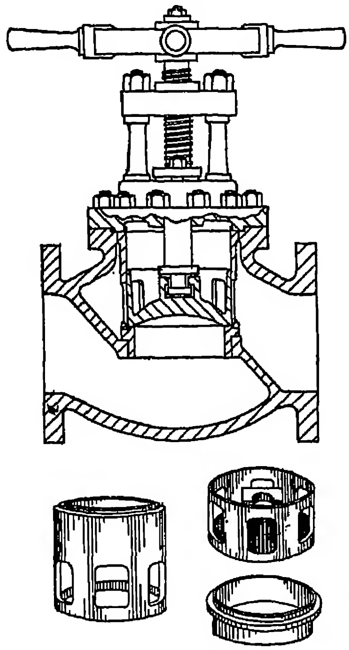


Fig. 68

Frequently a valve has to be placed in an inaccessible position, where it cannot be operated by an ordinary hand-wheel. This difficulty may be overcome by adopting either extended spindles and shafting fitted with bevel wheels, a sprocket wheel and chain, or a worm and worm wheel.

One of the figures shown illustrates a valve automatically operated, being controlled not by a screwed spindle but in some way by the fluid or working substance. The lift of a valve should not exceed one-fourth of its diameter, for with this lift the area of opening is equal to the area of inlet passage, so any further lift is useless. The amount of lift is regulated by a stop (see fig. 66).

Pump valves and check valves may be classed with such automatic valves.

Double-beat Valves.—The feature of this type of lift valve is that by the adoption of two mitres or faces, and two seatings, the lift of valve

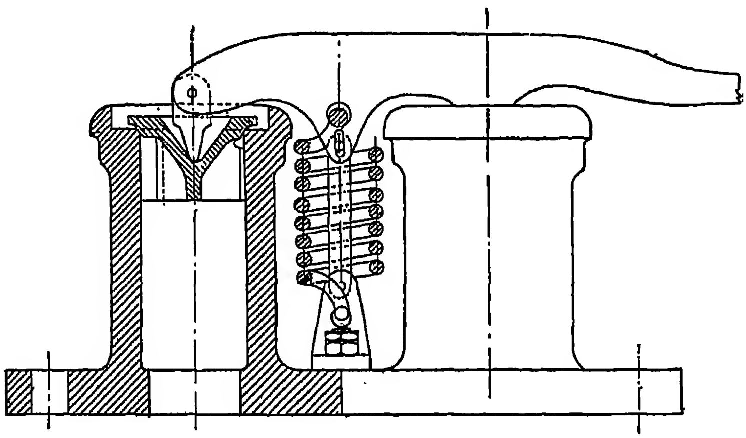


Fig. 69

may be considerably reduced without restricting the area of opening, and for valves of large diameter this is a distinct advantage.

Safety Valves.—In the marine and Ramsbottom type, the valves are

set to lift at a certain pressure against the pressure of helical springs. When the boiler pressure is normal, and not above the maximum allowed, the compression of the springs holds the valves on their seats, but once this pressure is exceeded the valves should open and allow steam to escape to waste until normal pressure is again restored. It is desirable to have as small a lift as possible, and, to meet this condition, valves of fairly large area are necessary; a practical refinement is gained by shaping the under surface of the valve as shown in fig. 69. The springs used are generally long, so that the compression is at a minimum for a small excess boiler pressure.

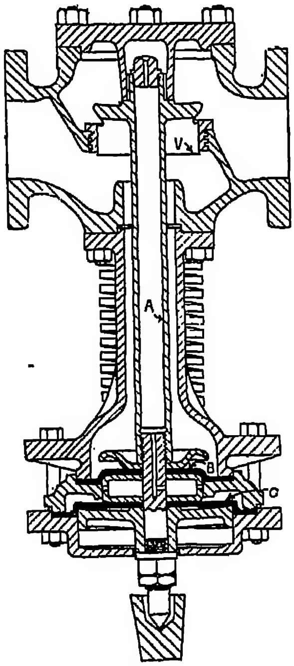


Fig. 70

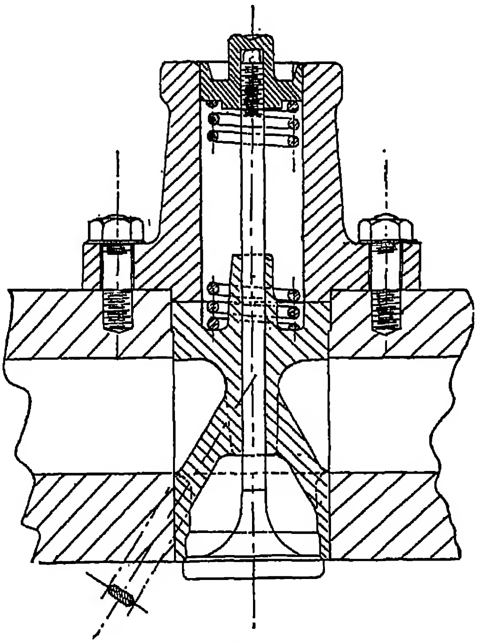


Fig. 71

The lever safety valve has a lever and weight in place of springs, and by sliding the weight along the lever, the position may be found where the valve will open for an increase in boiler steam-pressure.

The following simple formula may be used to find the distance from the fulcrum to the weight.

$$L = \frac{PAX}{W}, \text{ or } P = \frac{WL}{AX},$$

where P = pressure at which valve will blow off, in pounds per square inch,

A = area of valve opening in square inches,

X = distance from fulcrum to valve centre in inches,

W = weight of hanging mass in pounds,

L = distance from fulcrum to weight in inches.

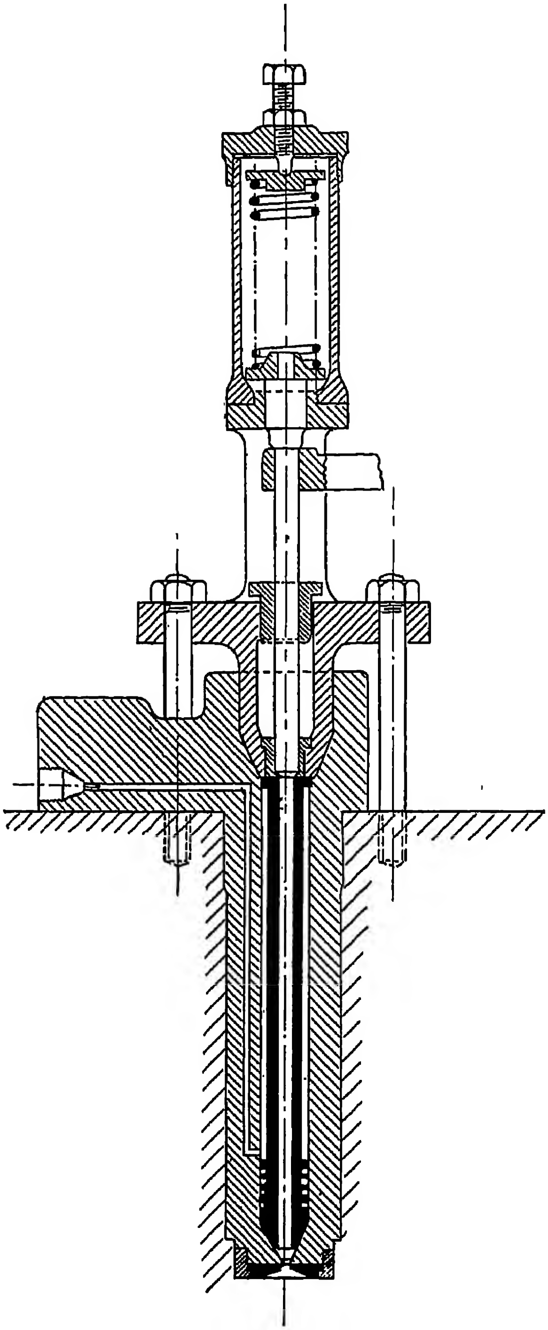


Fig. 72

The weight of the valve itself and the lever must also be taken into account when making the calculation.

Reducing Valves.—By the adoption of these valves high steam pressures are reduced automatically at one operation, and the supply of low-pressure steam passing can be regulated. There are many types on the market, one form having a piston and weighted lever, the piston being connected to a double-beat equilibrium valve. The pressure on the piston is counterbalanced by the load, due to the weighted lever, and on the pressure increasing the piston would be raised and thus close the valve.

Another type of valve giving excellent results is that manufactured by Messrs. Dewrance & Co. A sectional view of this valve is shown in fig. 70. Here the inlet pressure acts on the under side of valve V, and at the same time on the upper side of diaphragm B. These surfaces are exactly balanced, hence variation of the inlet pressure will not produce any effect on the outlet pressure. The reduced pressure on the upper side of valve V travels down the hollow valve spindle and acts on the under side of diaphragm B. The reduced pressure also acts on the upper side of diaphragm C, and it is arranged that this downward force exerted is balanced by the power

of the springs. Obviously, any increase in the reduced pressure will result in additional pressure on diaphragm C, which compresses the springs and closes the valve V.

Valves for Heavy-oil Engines (4-cycle).—The type of valve and casing usually adopted for the air inlet or the exhaust outlet is shown in fig. 71. It will be seen that the valve is of the mushroom type with guidance in the casing. The casing itself is of cast iron, with ports cast in its sides, and is arranged to fit into pockets in the cylinder cover. The valve is operated by a cam through the fulcrum lever against the compression of a strong helical spring.

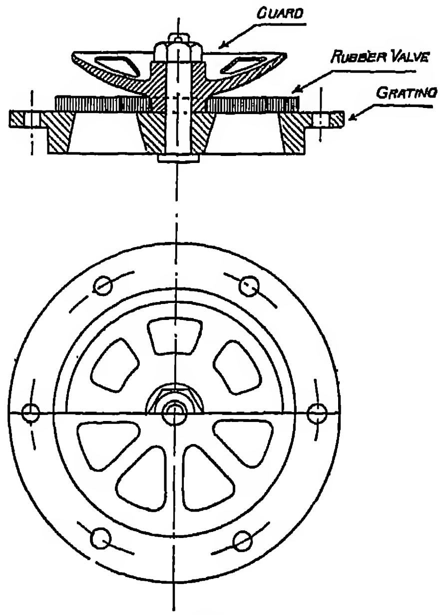


Fig. 73

Due to the high temperature of the exhaust gases the exhaust valve becomes very hot, so frequently these valves are made water-cooled internally. The inlet and outlet for the water are arranged at the top of the valve stem and holes drilled throughout its whole length, but this system is rather elaborate and liable to choke easily. Experience shows that a very economical valve, from the point of view of upkeep and renewals, is one with a cast-iron renewable end, fitted to a spindle of 3-per-cent nickel steel, to which, when the face of the valve is badly worn, a new detachable cast-iron end may be readily fitted.

The fuel inlet valve for Diesel engines is of complicated design, and is also operated from a cam. In addition to controlling the period of fuel injection to the combustion chamber, this valve is made to atomize the fuel while it is sprayed into the cylinder. A section of a typical Diesel engine fuel valve is illustrated in fig. 72.

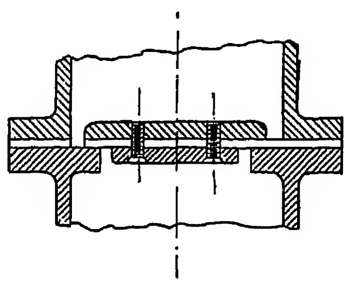


Fig. 74

Disc Valves.—These valves are of india-rubber or Dexine about ⅝ in. thick, and are used extensively for air pumps. The rubber or Dexine seats on a grating, and the valve opening or lift is fixed by a dished-shape guard, usually of spherical form. Two views of this type of valve are shown in fig. 73. Disc valves are also made of brass and even of steel.

Hinged Valves.—Valves of this class are called flap, clack, retaining.

or reflux valves. A simple flap valve is illustrated in fig. 74. The section shows the construction clearly. The valve itself consists of a piece of leather stiffened by plates (steel or gun-metal) and hinged at one side.

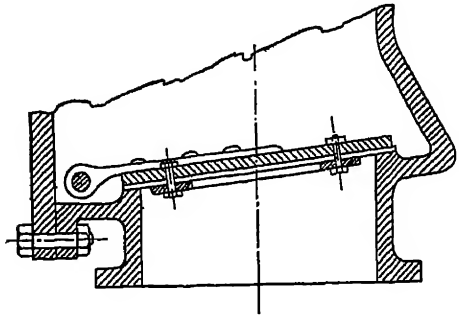


Fig. 75

The reflux valve shown in fig. 75 is more elaborate, the seating, which is scraped true, being part of the main casting. The valve is of leather and bolted between two plates, and the hinged bolt is so arranged that it can be operated externally.

CHAPTER IX

Gearing—Belt, Rope, and Tooth

Belt Gearing.—This extremely simple method of transmitting power, as from one machine to another, is extensively used in factories, engineering workshops, &c. Contrasted with a drive by toothed gearing, it is practically noiseless. When adopting a belt drive the following points are well worth notice.

Wherever possible make the lower side of belt the driving side, as the sag on the upper side will increase the arc of contact.

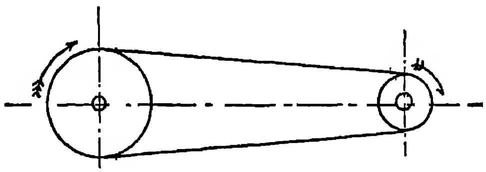


Fig. 76

If possible, arrange for horizontal drive, as a belt arranged vertically has not the effective grip as stated above.

Where any two pulleys are connected, the proportion of their diameters should not exceed a ratio of 6 to 1. If it is desired to increase the power of an existing drive without altering the shafting speed, the diameter of the pulleys should be increased, retaining the same ratio.

Belts over 18 in. wide are not to be recommended.

Connected shafts should not be arranged too closely, as the transmitted motion may be irregular, and for very long drives it is advisable to introduce guide pulleys to ensure a steady drive. The difference in tension between the upper and lower surfaces of a belt brings about what is termed " creep " or " slip " in the belt. Allowance for from 1 to 3 per cent lost motion due to this should be made.

Figs. 76–79 illustrate sundry arrangements of belt drives encountered in practice. It will be seen that in cases where the axes of the shafts are not parallel, the belt is led over guide pulleys.

The coefficient of friction with leather belting and cast-iron pulleys = 0·42.

Speed of Belts. — The speed of belting for main drives should not generally exceed 3500 to 4000 feet per minute, though often this does occur.

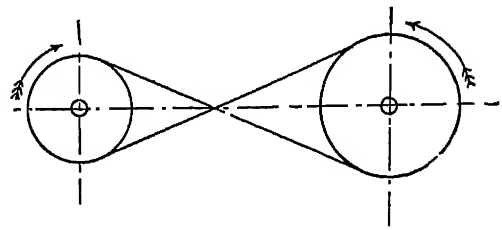


Fig. 77

Strength of Leather Belting. — The ultimate strength of leather belts varies considerably according to the tannage; ordinary leather belting may be assumed to have an average ultimate strength of about 4000 lb. per square inch, while chrome-tanned leather and rawhide have ultimate tensile strengths of about 5000 and 6500 lb. per square inch respectively.

Ordinary calculations relating to single leather belts are based on the following equation.

$$P = \frac{\text{H.P. } 33{,}000}{V},$$

where V = velocity in feet per minute,
P = driving pull in pounds,
H.P. = horse-power.

Thus it will be seen that, having first determined the driving pull, by dividing this by the safe tension allowable per inch of width, the width of belt required may be calculated. A safe figure to allow for the tension of belting is 50 lb. per inch of width for a single leather belt, or 80 lb. per inch of width for double leather belting.

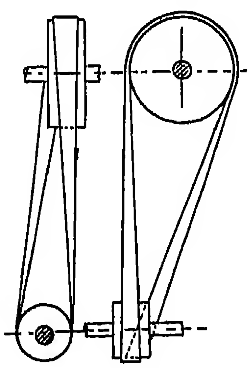


Fig. 78

The average thickness of leather belting is $\frac{7}{32}$ in.

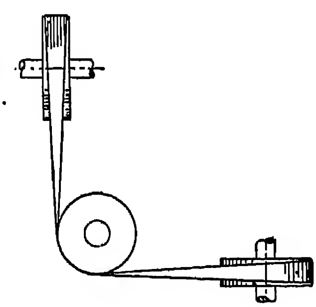


Fig. 79

Rope Gearing.— This system has found much favour in the Lancashire district for driving textile machinery.

Rope gearing may be considered under two headings, namely, the English or individual rope drive, and the American or continuous drive. In the former, the driving pulley, generally the fly-wheel

of the prime mover, has a number of grooves machined on its periphery, each groove carrying a single rope which also passes over a similarly grooved pulley, thus transmitting power to machinery in various parts of the factory. With the continuous system, one long rope is arranged to drive all the pulleys, and guide pulleys are used to adjust the tension in the rope. The disadvantage of this arrangement is that, in the event of the rope breaking, the whole plant will be brought to a standstill.

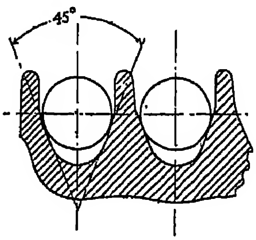


Fig. 80

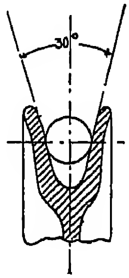


Fig. 81

To ensure good results the grooves in the pulleys must be of wedge section, an angle of 30° being usually adopted for small ropes under 1 in. diameter, and of 45° for larger sizes, and care must be taken that all grooves are of standard form. Figs. 80 and 81 illustrate two types of groove.

It has been found that ropes of hemp and manila do not wear as well as cotton ropes.

It is customary to run rope drives at a speed averaging 4500 ft. per minute, but with speeds much in excess of this figure the centrifugal tension set up, which decreases the actual working tension of the rope, must be taken into account. This may be stated thus:

$$D = \frac{wv^2}{32},$$

where w = weight per foot length of rope,
v = velocity in feet per second,
D = centrifugal tension in pounds.

Taking into consideration this centrifugal tension, a safe average figure for the stress = 200 lb. per square inch of section.

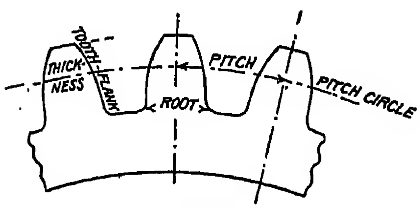


Fig. 82

It is recommended that the diameter of a rope pulley be not less than 30 times the rope diameter (the diameter of pulley being measured between the centres of the rope), or the life of the rope will be reduced by the severe bending action caused through a pulley being of small diameter.

Toothed Gearing.—By reference to fig. 82 the technical terms used for toothed gearing will be clearly seen. By circular pitch is meant the length of the arc measured on the pitch circle from the centre of any tooth to the centre of the adjacent tooth. Then

$$Np = \pi D \text{ or } D = \frac{Np}{\pi},$$

where N = number of teeth,
p = pitch in inches,
D = diameter of pitch circle in inches.

The diametral pitch may be found by dividing the number of teeth by the pitch circle diameter in inches. The outside diameter of a spur gear

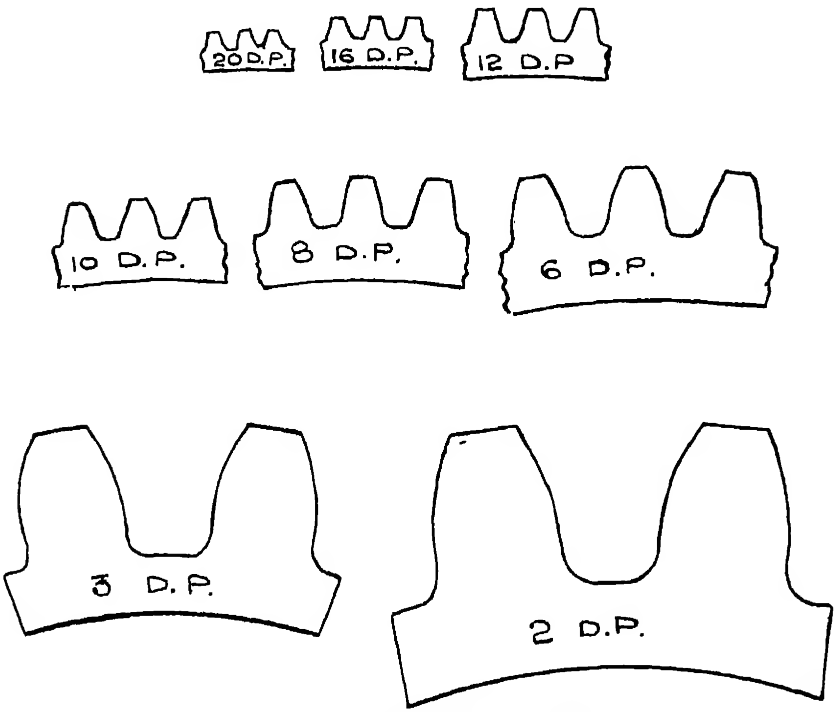


Fig. 83.—Comparative Sizes of Involute Teeth

may be found by adding 2 to the number of teeth and dividing by the diametral pitch. Different authorities give various proportions for teeth as specifically suited to different classes of work, possibly with divergent views on economy in manufacture. For ordinary work the proportions in terms of the circular pitch may be taken as follows:

Thickness of tooth	= 0·5	pitch.
Width of space between teeth	= 0·5	„
Length of tooth above pitch line	= 0·32	„
Length of tooth below pitch line	= 0·36	„
Width of tooth	= 2 to 3	„

To ensure smooth-running conditions with uniform motion attention must

be paid to the tooth formation. Two curves meeting these requirements are cycloid and involute, and are adopted for the formation of wheel teeth.

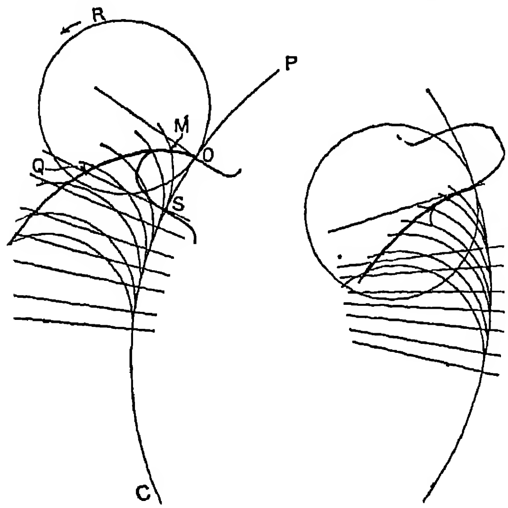


Fig. 84.—Tooth Flank—Cycloidal Type

Cycloidal Tooth.—Having first fixed the pitch circle diameter the flank of the tooth can be determined by rolling circles. The flank is formed by a hypocycloid and the face by an epicycloid curve.

Let PC be the pitch circle of the driver-wheel (fig. 84) and R the rolling circle. Then a point on the epicycloid is easily found from the fact that the *arc* SO equals the arc SM, i.e. $R\theta = r\phi$, where R is the radius of the pitch circle, r of the rolling circle, and θ and ϕ are the angles subtended by the respective arcs at the centres. The rolling circle is drawn in any displaced position, θ, and the radius QM is drawn so that $\angle$SQM equals $(\theta.R/r)$. M then lies on the epicycloid. The hypocycloidal form for the flank of the tooth on the follower-wheel is found similarly, the only essential difference being that the rolling circle now rolls *inside* the pitch circle of the follower-wheel.

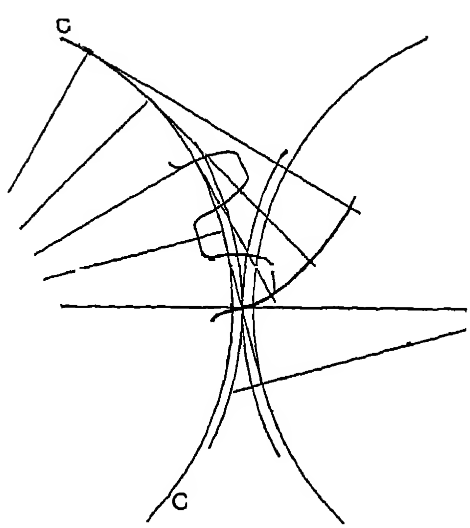


Fig. 85.—Tooth Flank—Involute Type

Involute Tooth.—The involute of the base circle C is shown in fig. 85. To construct this curve draw a straight line equal in length to one-half of the circumference of the base circle and divide it into any number of equal parts. Divide the semicircle into a like number of parts and draw tangents to the radial division lines as shown. On a tangent mark lengths corresponding to the divisions found on the "half-circumference line". A line drawn through these points will form the involute.

Strength of Teeth.—When calculating the strength of a tooth, one tooth only of each wheel is assumed to engage at any instant. The power transmitted is represented

as a force P acting at the pitch circle diameter of the wheel, and the tooth may be considered as a cantilever loaded at one end. Two distinct cases may be considered, firstly, with the load uniformly distributed throughout the entire width of tooth, and secondly, where the load or pressure acts on one corner of the tooth only (see figs. 86 and 87). In the case of

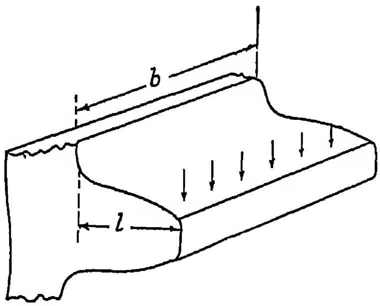


Fig. 86

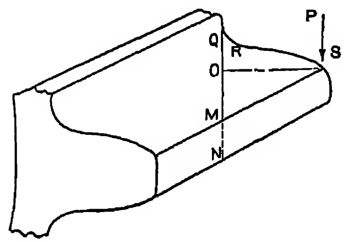


Fig. 87

the tooth uniformly loaded, by equating the maximum bending moment to moment of resistance to bending then

$$Pl = \frac{bt^2}{6} f, \text{ or } f = \frac{6Pl}{bt^2},$$

where P = pressure on tooth in pounds,
l = length of tooth,
f = stress in pounds per square inch,
$\frac{bt^2}{6}$ or Z = Modulus of section (assuming the tooth root to be of rectangular section).

The load on tooth P is readily calculated from the horse-power absorbed, thus

$$P = \frac{\text{H.P.} \times 33000 \times 12}{2\pi RN},$$

where H.P. = horse-power absorbed,
R = radius of pitch circle of wheel in inches,
N = revolutions per minute.

If the pressure be considered as acting at a corner of the tooth (see fig. 87), then the tooth would tend to break at section MNQR at an angle of 45°. Here $Pl = Zf$ as before, where l is equivalent to distance OS on diagram, and the modulus $Z = \frac{bt^2}{6}$, where $b = 2\text{OS}$.

The method, however, generally employed for determining the strength of gears, particularly cut gears, is that introduced by Mr. Wilfrid Lewis (Engineers' Club of Philadelphia, 1892). Here consideration is given to the tooth formation and the safe working stress at a given speed. This

formula, known as the Lewis formula, may be expressed as follows:

$$w = fpBy,$$

where w = tangential load in pounds,
f = safe stress at given velocity,
B = width of face,
y = outline factor,
p = circular pitch.

It should be noted that the safe stress is dependent on the material and the speed, therefore the following tables should be used in conjunction with the above formula.

TABLE I.—y OUTLINE FACTOR (CIRCULAR PITCH).—W. LEWIS

Number of Teeth.	Involute 20° Obliquity.	Involute 15° and Cycloidal	Radial Flanks.
12	0·078	0·067	0·052
13	0·083	0·070	0·053
14	0·088	0·072	0·054
15	0·092	0·075	0·055
16	0·094	0·077	0·056
17	0·096	0·080	0·057
18	0·098	0·083	0·058
19	0·100	0·087	0·059
20	0·102	0·090	0·060
21	0·104	0·092	0·061
23	0·106	0·094	0·062
25	0·108	0·097	0·063
27	0·111	0·100	0·064
30	0·114	0·102	0·065
34	0·118	0·104	0·066
38	0·122	0·107	0·067
43	0·126	0·110	0·068
50	0·130	0·112	0·069
60	0·134	0·114	0·070
75	0·138	0·116	0·070
100	0·142	0·118	0·071
150	0·146	0·120	0·072
300	0·150	0·122	0·073
Rack	0·154	0·124	0·075

TABLE II.—SAFE STRESS f FOR DIFFERENT SPEEDS

Velocity in Feet per Minute.	100 or Less.	200	300	600	900	1200	1800	2400
Steel ..	20,000	15,000	12,000	10,000	7500	6000	5000	4300
Cast iron	8,000	6,000	4,800	4,000	3000	2400	2000	1700

Bevel Gearing.—Motion may be transmitted between shafts whose axes intersect by means of bevel gears. Gears of equal size connecting two shafts at right angles are commonly called mitre wheels.

The following rules indicate the manner in which the proportions of bevel gears may be calculated, where the shafts carrying the gears are at right angles.

The pitch cone angles of pinion and gear (see fig. 88, A) are drawn to agree with the desired speed ratio. The pitch cone angle of the pinion may be found by dividing the number of teeth in the pinion by the number of teeth in the gear, this giving the tangent of the angle required. Similarly, the tangent of the pitch cone angle of the gear may be found by dividing the number of teeth in the gear by the number of teeth in the pinion. As a proof of these calculations the sum of the two angles should equal 90°.

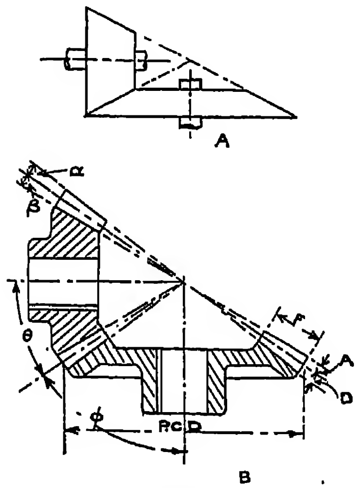


Fig. 83

Assuming the teeth to be machine cut, the addendum and dedendum are obtained by dividing 1 and 1·157 respectively by the diametral pitch. This amount, it should be noted, is set off at the pitch cone radius, this latter being equal to one-half the pitch circle diameter multiplied by the co-secant of the pitch cone angle (see fig. 88, B).

The tangent of the addendum and dedendum angles may be obtained by dividing the addendum and dedendum respectively by the pitch cone radius. The foregoing may be more conveniently expressed as follows:

where DP = diametral pitch,
A = addendum,
D = dedendum,
NP = number of teeth in pinion,
NG = number of teeth in the gear,
Tan θ = tangent of pitch cone angle of pinion,
Tan ϕ = tangent of pitch cone angle of gear,
Tan α = tangent of addendum angle,
Tan β = tangent of dedendum angle,
PCD = pitch circle diameter,
CR = pitch cone radius,
F = width of bevel face.

Thus

1. $$\text{Tan } \theta = \frac{\text{NP}}{\text{NG}}.$$

2. $$\text{Tan } \phi = \frac{\text{NG}}{\text{NP}}$$

3. $$\text{A} = \frac{1}{\text{DP}}.$$

4. $$\text{D} = \frac{1 \cdot 157}{\text{DP}}.$$

5. $$\text{CR} = \frac{\text{PCD}}{2} \times \text{cosecant of pitch cone angle.}$$

6. $$\text{Tan } \alpha = \frac{\text{A}}{\text{CR}}.$$

7. $$\text{Tan } \beta = \frac{\text{D}}{\text{CR}}.$$

8. F may be taken as 2·6 times the circular pitch.

Strength of Bevel Gears.—This may be calculated, using the Lewis formula, and taken as a spur gear of the same number of teeth, pitch, and face. To compensate for the varying section of tooth it is necessary to multiply the result by $\frac{d}{\text{D}}$ where d = small pitch diameter and D = large pitch diameter.

Worm Gearing.—Reduction from high speeds with a limited number of parts may be accomplished by worm gearing. It has been found that the best results are obtained by using a hardened-steel worm engaging a hobbed phosphor-bronze worm wheel. The worm should be arranged to revolve in an oil bath on the under side of gear, while the adoption of both radial and thrust ball bearings adds considerably to the efficiency of the drive.

Multiple-threaded worms are largely used, as they permit worm drives of small reduction; again, worms of this type give an increased angle of thread or helix angle, this being important as the efficiency of the worm gearing increases as the helix angle approaches 45°.

Distinction should be made between the terms "lead" and "pitch", although for a single-threaded worm they are equal. With a double-threaded or two-start worm, the lead is equal to twice the pitch, and for a triple-threaded, or three-start worm, three times the pitch.

The standard form of worm tooth is similar to the involute rack tooth, where the angle of the tooth side is inclined at 14½° to the vertical. By reference to fig. 89 the terms used in connection with worm gearing may be seen.

When designing worm gearing the diameter of the worm should be kept

as small as possible, and is usually selected with a view of utilizing a standard hob already in stock.

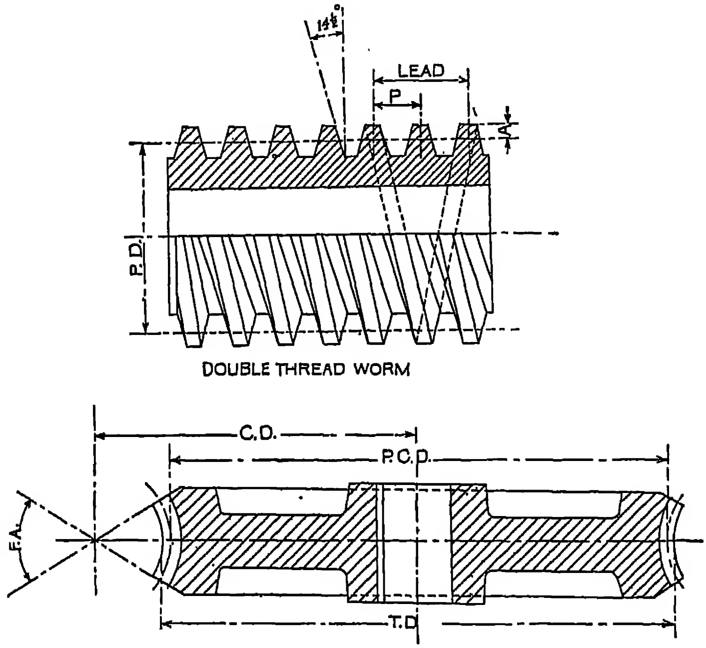


Fig. 89

The proportions of the worm and wheel may be derived from the following:

where N = number of teeth in worm wheel,
P = linear pitch of worm,
A = addendum of worm tooth,
PD = pitch diameter of worm,
PCD = pitch circle diameter of worm wheel,
TD = throat diameter of worm wheel,
CD = centre distance between worm and wheel,
FA = face angle of worm wheel.

Thus

1. $$A = P \times 0{\cdot}318.$$

2. $$PCD = \frac{N \times P}{3{\cdot}1416}.$$

3. $$CD = \frac{PCD + PD}{2}.$$

4. $$TD = PCD + 2A.$$

5. FA may be taken between 60° and 80°.

Helical Gearing.—This form of gearing is considerably stronger than ordinary spur gearing. In the double-helical type the teeth are V shaped, this counteracting any tendency of the gears to separate axially. To ensure that the pressure on the teeth is equally distributed, one of the gears is arranged to have a small amount of end play.

In turbine double-reduction gears for steamships, the end thrust is counteracted by adopting oppositely cut helices; here high-pressure turbine revolutions up to 3000 r.p.m. are reduced to a propeller-shaft speed of about 100 r.p.m.

Non-metallic Pinions.—Where noiseless running is desired, pinions made of raw hide, cotton fibre, &c., are generally used. Pinions made of these materials possess shock-absorbing qualities and prevent the transmission of vibration. It is claimed by manufacturers that pinions made of fibre, compressed paper, &c., are unaffected by temperature changes and do not deteriorate under the action of acid fumes, oil, and water.

Construction.—Cotton fibres are compressed under hydraulic pressure, and are held in compression by steel plates, commonly called shrouds. Screwed rivets passing through both fibre and shrouds complete the assembly. After machining, the blanks are impregnated with oil and again after the teeth are cut.

It is important that the companion gear should engage the fibre portion of the pinion only, therefore sufficient clearance for the steel shrouds should be provided.

TABLE OF CIRCULAR PITCHES WITH EQUIVALENT DIAMETRAL PITCHES

Circular Pitch.	Diametral Pitch.	Circular Pitch.	Diametral Pitch.
Inches.	Inches.	Inches.	Inches.
2	1·571	$\frac{7}{8}$	3·590
$1\frac{7}{8}$	1·676	$\frac{13}{16}$	3·867
$1\frac{3}{4}$	1·795	$\frac{3}{4}$	4·189
$1\frac{5}{8}$	1·933	$\frac{11}{16}$	4·570
$1\frac{1}{2}$	2·094	$\frac{5}{8}$	5·027
$1\frac{7}{16}$	2·185	$\frac{9}{16}$	5·585
$1\frac{3}{8}$	2·285	$\frac{1}{2}$	6·283
$1\frac{5}{16}$	2·394	$\frac{7}{16}$	7·181
$1\frac{1}{4}$	2·513	$\frac{3}{8}$	8·378
$1\frac{3}{16}$	2·646	$\frac{5}{16}$	10·053
$1\frac{1}{8}$	2·793	$\frac{1}{4}$	12·566
$1\frac{1}{16}$	2·957	$\frac{3}{16}$	16·755
1	3·142	$\frac{1}{8}$	25·133
$\frac{15}{16}$	3·351	$\frac{1}{16}$	50·266

TABLE OF DIAMETRAL PITCHES WITH EQUIVALENT CIRCULAR PITCHES

Diametral Pitch.	Circular Pitch.	Diametral Pitch	Circular Pitch.
Inches.	Inches.	Inches.	Inches.
$1\frac{1}{4}$	2·5133	11	0·286
$1\frac{1}{2}$	2·0944	12	0·262
$1\frac{3}{4}$	1·7952	14	0·224
2	1·571	16	0·196
$2\frac{1}{4}$	1·396	18	0·175
$2\frac{1}{2}$	1·257	20	0·157
$2\frac{3}{4}$	1·142	22	0·143
3	1·047	24	0·131
$3\frac{1}{2}$	0·898	26	0·121
4	0·785	28	0·112
5	0·628	30	0·105
6	0·524	32	0·098
7	0·449	36	0·087
8	0·393	40	0·079
9	0·349	48	0·065
10	0·314		

RIVETS AND RIVETED JOINTS

BY

ANDREW S. LINDSAY

Rivets and Riveted Joints

Rivets are such an important factor in mechanical engineering that great care should be taken in the selection of the material from which they are made; also in their formation and the manner in which they are " closed " or riveted.

Formerly a soft ductile iron with a tensile strength of 23 tons per square inch and elongation of 25 per cent was extensively used, but this has now given place to a low-carbon steel with a tensile strength of 23 to 25 tons per square inch with an elongation of 30 per cent, the steel rivet being freer from laminations. The iron rivet, however, is still sometimes specified where hand riveting is necessary, as in awkward corners and positions inaccessible for a riveting machine. The snap, or cup-headed rivet, fig. 1, is the one used in machine riveting, and fig. 2 shows a hand rivet; other figures, 3 to 6, being modifications of these. Fig. 3 is a hand rivet with full countersunk head, and fig. 4 a flush countersunk rivet also hand riveted. Fig. 5 is a machine rivet, the snap part being countersunk. Fig. 6 is similar to fig. 5, except that the head in this case is called a half snap.

All rivets should be machine riveted where possible, as this ensures the rivet being made to fill the hole better, and therefore take a better shear strain. It also closes the plates together, and as each rivet gets the *same pressure* there is no danger of slack rivets. Where flush rivets are used the countersink may vary, and it may be difficult to judge the exact amount of material required to fill each hole. The rivet should therefore be left " full " and chipped flush with the plate. It is advisable in dealing with long rivets, which, owing to excessive contraction, sometimes draw the heads off, that the heads should be water-cooled before placing the rivet into the hole for riveting.

Rivet Holes.—All holes in boiler work should be drilled in the plates after they have been bent to shape and placed in position so as to ensure perfect alignment. Punching the holes seriously damages the plates round about the holes, setting up small cracks which may develop later on when under working conditions. Punched holes never agree in spacing or alignment, and have to be forced into their proper relation by drifting, that is, driving a tapered mandrel into the holes, thereby forcing them into line. This injury to the plates can only be removed by subsequently annealing them, or drilling the holes larger, thus removing the damaged material. For these reasons this practice is prohibited in the best boiler practice.

Types of Riveted Joints.

Single-riveted Lap Joint.—When the plates to be riveted together overlap one another and are united by a single row of rivets, the joint, fig. 7, is termed a single-riveted lap joint. When such a joint is put in tension, a bending moment is set up at the rivet, due to the plates not being in line with each other, and subsequently they are subjected to additional stress, thereby weakening the joint. The strength of this form of joint is approximately 54 per cent that of the unweakened plate, and is only adopted for small boilers and steam drums with *low* pressures, where the plates are made thicker than is necessary for the strength actually required. To ensure single-riveted lap joints being steam-tight the lap given to the plates is three times the diameter of the rivet, and when made either more or less than this, fullering difficulties arise; the plates spring apart, or tear at the edges.

Double-riveted Lap Joint.—In this case the overlapping plates are united together by a double row of rivets, either by " zigzag " or " chain " riveting. In the former method, fig. 8, the pitch centres of one row are opposite the middle of the pitch centres of the other row, and in the latter method, fig. 9, the pitch centres of each row are in line, the weakening of the plates in this case being less than, but the joint not so steam-tight as in zigzag riveting. A good average strength for double-riveted lap joints is about 66 per cent that of the solid plate, and by the addition of a third row of rivets, with, of course, a corresponding increase in lap, this strength may be raised to 72 per cent, the joint now formed, fig. 10, being termed a *treble-riveted lap joint.*

Single-riveted Single-butt Strap Joint.—In a butt joint the plates are in the same plane and are joined together by a butt strap which is riveted to each plate by a single row of rivets. On the application of a pull to the plates the same bending action takes place as in a single-riveted lap joint, with the same effects. This form of joint, fig. 11, is seldom, if ever, employed now in boiler work.

Single-riveted Double-butt Strap Joint.—A joint of this type, fig. 12, consists in having two butt straps, one outside and one inside, there being a single row of rivets through each plate. The strength of this joint for steam-tightness is about 60 per cent that of the solid plate, but may be raised to perhaps 70 per cent on making the butt straps the same thickness as the plates. The bending moment on the plates is eliminated by having the two butt straps instead of one, the rivets, however, being in double shear.

Double-riveted Double-butt Strap Joint.—When double-riveted, fig. 13*a*, the strength of the joint may be increased to 75 per cent that of the unweakened plate, and is extensively used when the boiler plates are thick. A still further increase in the strength of this joint can be obtained by the omission of alternate rivets in the outer rows, leaving the two inner rows with double the number of rivets of the two outer, as in fig. 13*b*. The strength in this case may be as high as 81 per cent.

Treble-riveted Double-butt Strap Joint.—In this form, fig. 14*a*, there are three rows of rivets through each plate, i.e. six rows through the two butt straps. The strength of the joint is the same as that of a treble-riveted lap joint with the middle row of rivets twice that of the two outer. When the

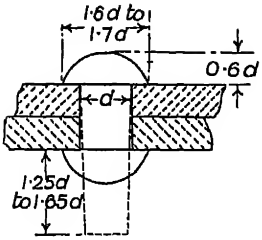


Fig. 1

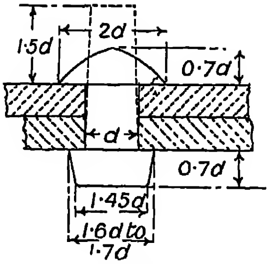


Fig. 2

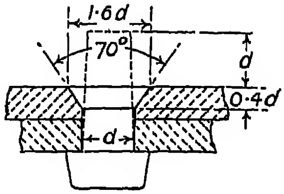


Fig. 3

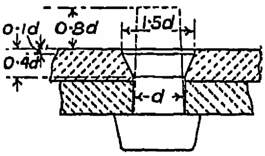


Fig. 4

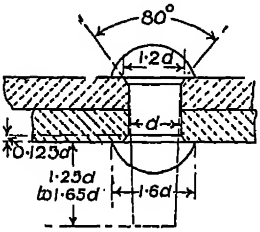


Fig. 5

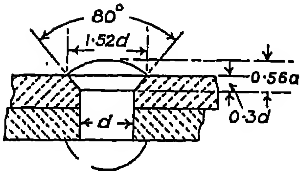


Fig. 6

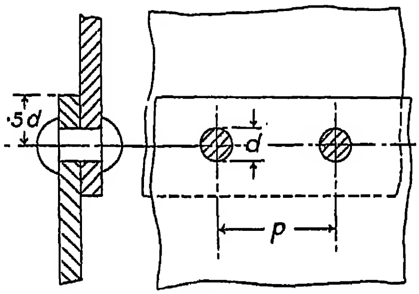


Fig. 7

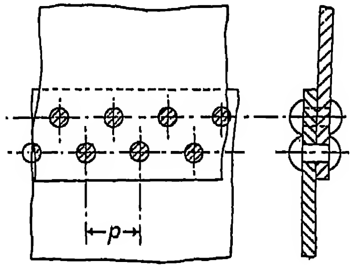


Fig. 8

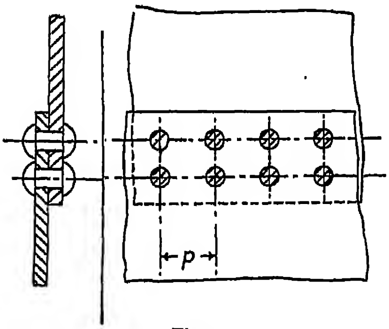


Fig. 9

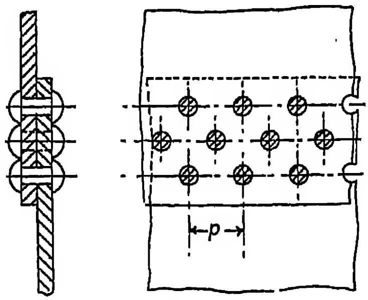


Fig. 10

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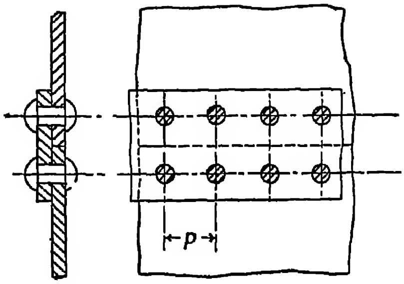

Fig. 11

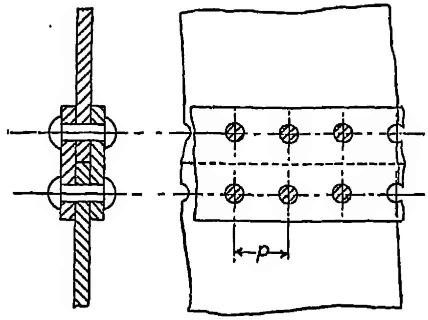

Fig. 12

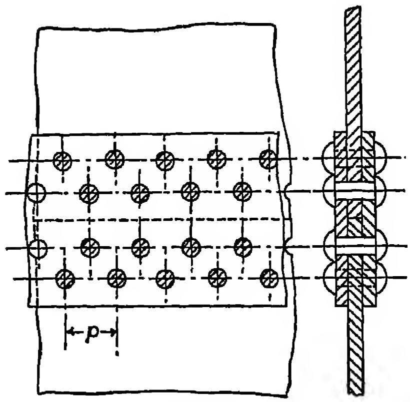

Fig. 13a

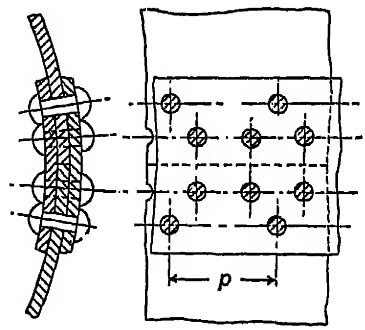

Fig. 13b

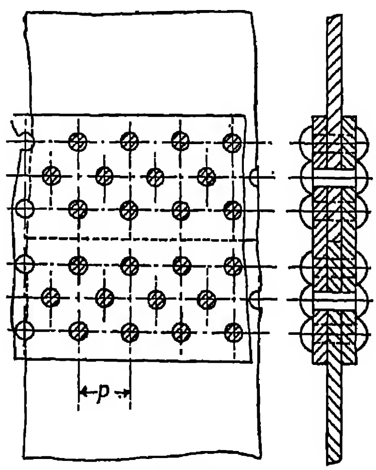

Fig. 14a

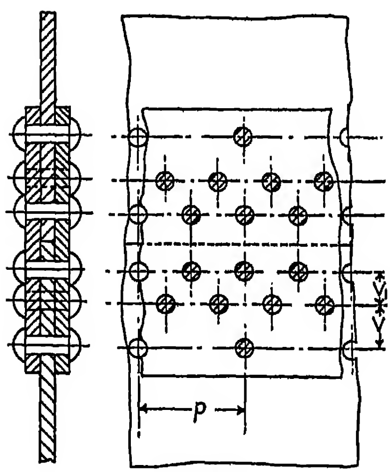

Fig. 14b

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rivets in the four inner rows of a treble-riveted butt joint are twice that of the two outer, the joint has a strength of 85 per cent that of the solid plate. The difficulty of fullering a joint of this description, fig. 14*b*, is the factor which prevents its application to boiler work except when the plates are thick, say not under 1 in.

Combination Lap Joint.—Fig. 15 illustrates a single-riveted combination lap joint with a single cover plate or butt strap. The central row of rivets passes through the two plates and the butt strap, but has twice the number of rivets that are in either of the two outer rows. The strength of the joint is about 82 per cent that of the solid plate, but is further increased by replacing the middle row of rivets by two rows, each again having twice the number of rivets than in either the two outer. This joint is now known as a double-riveted combination lap joint with a single butt strap as in fig. 16.

The types of joints commonly adopted for longitudinal seams in boilers are the double-riveted and treble-riveted butt joints with double butt straps. Lap joints are seldom used for first class longitudinal joints owing to their liability to fracture with constant use. Circumferential or girth seams are generally single- or double-riveted lap joints, since the transverse bursting effort is considered to be only half that of the circumferential bursting effort.

Thickness of Butt Straps.—Theoretically the thickness of the butt straps should be half the thickness of the plates, since there are two of them. In practice the straps are made five-eight times the plate thickness, although frequently the one on the inside of the boiler seam is made a little thicker than the outer. In the case of a butt joint with alternate rivets omitted in the outer rows, the theoretical thickness of the straps is greater than half the thickness of the plates, because the strengthening effect gained by the omission of the rivet holes in the plates is absent in the straps. In practice the straps are generally designed in this case from the standard expression

$$t_1 = \tfrac{5}{8}\left(\frac{p-d}{p-nd}\right)t,$$

where p is the wide pitch, d the diameter of the rivet, t the thickness of the plate, t_1 the thickness of the butt straps, and n the number of inner-row rivets per wide pitch p. When diamond riveting is adopted and the plates, in the case of lap joints, or the butt straps in butt joints, are made of thin material, they are sometimes scalloped to facilitate fullering.

Structural Riveted Joints.—In figs. 17*a*, *b*, and *c* are illustrated three arrangements of riveting a tie-bar. In each case the two strips of plate are united by a butt strap on each side of the joint and fastened together by twelve rivets. As the number of rivets is the same in each case, they will of course have the same diameter.

Let t be the load on each rivet,
D, the width of the plate,
d, the diameter of the rivet.
A, the area of section considered,
p, the stress in the plate.

Consider fig. 17*a*.

At section AA, $\left(\frac{D - 3d}{D}\right)Ap_1 = 3t_1.$

" BB, $\left(\frac{D - 3d}{D}\right)Ap_2 = 6t_1.$

$$\therefore p_2 = \frac{TD}{(D - 3d)A}.$$

T is the total tension in the tie-bar.

Consider fig. 17*b*.

At section CC, $\left(\frac{D - 2d}{D}\right)Ap_3 = 2t_2.$

" DD, $\left(\frac{D - 2d}{D}\right)Ap_4 = 4t_2.$

" EE, $\left(\frac{D - 2d}{D}\right)Ap_5 = 6t_2.$

$$\therefore p_5 = \frac{TD}{(D - 2d)A}.$$

Consider fig. 17*c*.

At section FF, $\left(\frac{D - 3d}{D}\right)Ap_6 = 3t.$

" GG, $\left(\frac{D - 2d}{D}\right)Ap_7 = 5t.$

" HH, $\left(\frac{D - d}{D}\right)Ap_8 = 6t.$

$$\therefore p_8 = \frac{TD}{(D - d)A}.$$

Since $(D - d) > (D - 2d) > (D - 3d)$, then $p_2 > p_5 > p_8$; hence the last method, *c*, of uniting the plates is obviously the best of the three and is largely used in constructional work, but not so generally applicable in boiler work. It will be noticed that although the strength of the plate has been increased by adopting *c* instead of *a* or *b*, the butt straps are still cut away by a continuous row of rivet holes, therefore to bring their strength up to that of the plate their thickness must be increased in the ratio of $(2p - d)$ to $(2p - 2d)$. In actual practice they would each be made about three-quarters of the thickness of the plate.

In the case of a flat tie-bar, fig. 18, used as a bracing member of a roof truss, to ensure that the line of thrust passes through the centre of gravity of the section of the bar, the arrangement of the rivets must be such that they are symmetrical about the two centre lines of the members to be riveted.

Staggering of Rivets.—In constructional work such as the girders of bridges, the moment of inertia of the beam about the neutral axis is usually increased by riveting steel plates along the top and bottom flanges.

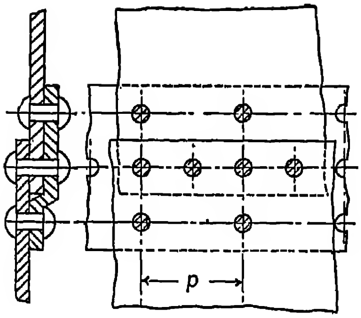

Fig. 15

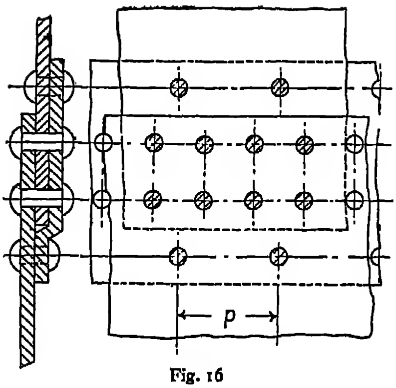

Fig. 16

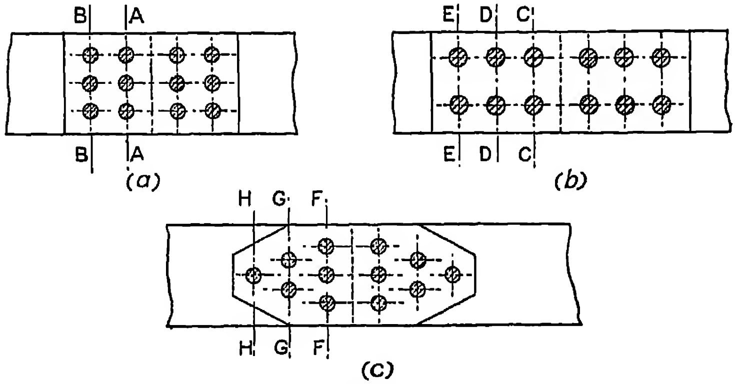

Fig. 17

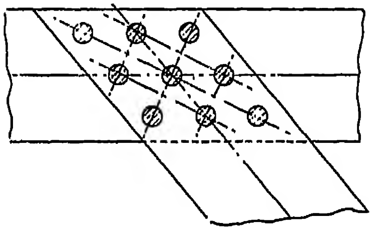

Fig. 18

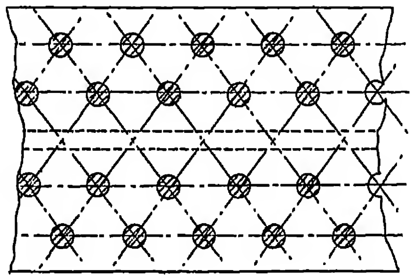

Fig. 19

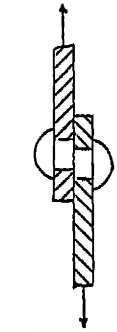

Fig. 20

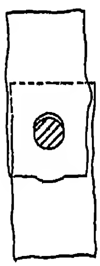

Fig. 21

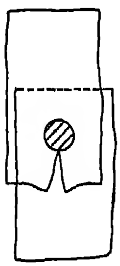

Fig. 22

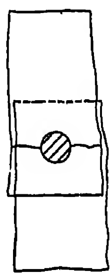

Fig. 23

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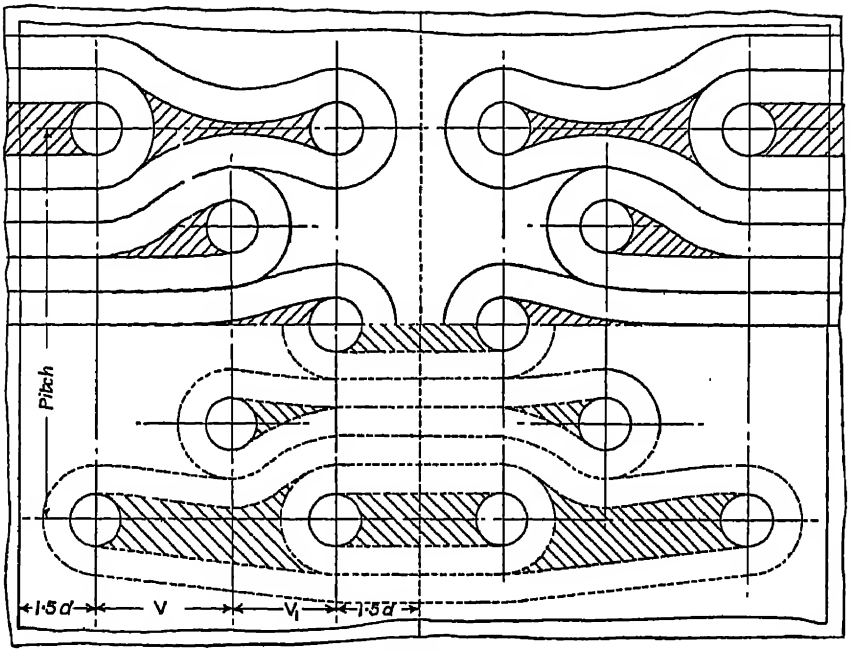

Fig. 24

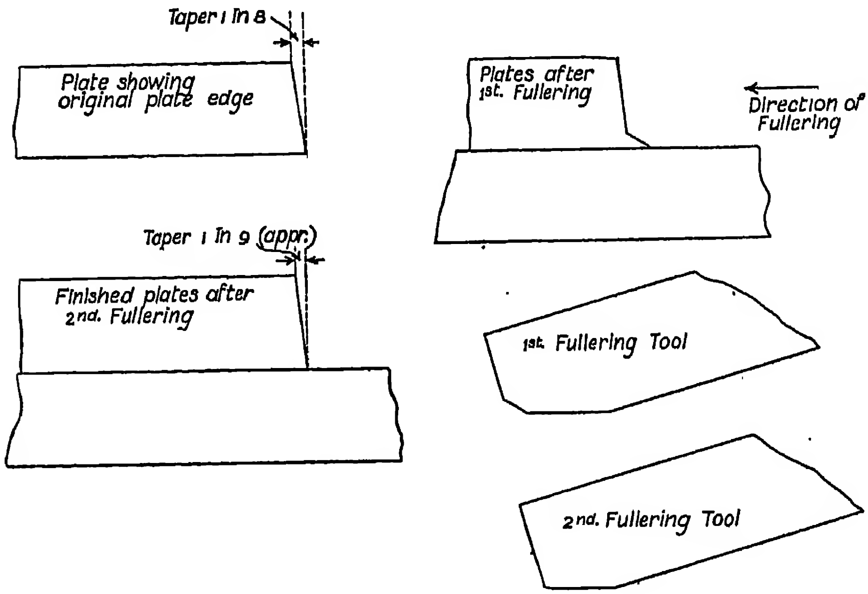

Fig. 25

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The rivets used are staggered as shown in fig. 19. The object of this is to avoid drilling more holes than is absolutely necessary in any particular cross section. The fewer holes there are the greater will be the moment of inertia about the neutral axis and the stronger the beam.

Failure of Riveted Joints.—A riveted joint may fail: (1) by the shearing of the rivet as in fig. 20; (2) by crushing of the plate, fig. 21; (3) by the tearing out of the rivet as in fig. 22; and (4) by the tearing of the plate across, fig. 23.

The ultimate resistance of a rivet to failure by shearing is $\frac{\pi d^2}{4} f_s$, where d represents the diameter of the rivet which is supposed to be in single shear and f_s is the ultimate shear stress of the rivet.

The ultimate resistance offered by the plate to crushing, i.e. the bearing pressure on the rivet, is dtf_B, t being the thickness of the plate and f_B the ultimate bearing pressure.

In the case of the joint failing by the tearing out of the rivet, a rough idea of what happens may be got by considering the plate in front of the rivet to be a free-free beam of length equal to the diameter of the rivet and loaded uniformly with a load equal to that on the rivet. This assumption is probably far from true, but it *under-estimates* the strength of the plate to tearing and hence is a *safe* assumption. For this reason the bending moment on the plate edge is taken to be $\frac{wl^2}{12}$ instead of $\frac{wl^2}{8}$. Let l be the distance of the edge of the plate from the nearest side of the rivet.

$$\text{Load on rivet} = \frac{\pi d^2}{4} f_s.$$

$$\therefore \text{ Bending moment on the plate} = \frac{\pi d^2}{4} f_s \times \frac{d}{12} \quad \ldots\ldots\ldots(1)$$

$$\text{Also moment of resistance of plate} = \tfrac{1}{6} t l^2 f_t \quad \ldots\ldots\ldots\ldots(2)$$

Equating (1) and (2), then

$$l^2 = \frac{\pi d^3 f_s}{8 t f_t} \quad \ldots\ldots\ldots\ldots\ldots\ldots(3)$$

Making the ultimate resistance to crushing the same as the ultimate resistance to shearing, then

$$f_s \frac{\pi d^2}{4} = f_B dt;$$

$$\therefore t = \frac{\pi d}{4} \frac{f_s}{f_B}.$$

Substituting for f_s in (3), then

$$l^2 = \frac{f_B d^2}{2 f_t},$$

$$\text{i.e. } l = d\sqrt{\frac{f_B}{2 f_t}}.$$

It is obvious that the above relation holds for butt straps when each is half the thickness of the boiler plate.

Considering the plate to tear along the line of rivets, the resistance to failure is Af_t where A is the effective sectional area along the rivet line.

The aggregate result of the foregoing will be seen to be that failure by shearing depends on the rivet diameter; failure by crushing the plates depends on diameter of rivet and thickness of plate; and that failure by tearing depends on the thickness of the plate and the width on each side of the rivet. Obviously to produce an ideal riveted joint these three strengths should be as nearly equal as may be, and in the design of such joints this equality of strength is the draughtsman's prime endeavour.

Design of Riveted Joints.—In the following work on riveted joints let:

d represent the diameter of the rivet,
p, the pitch of rivets,
t, the thickness of plates,
f_t, the working tensile stress in plates,
f_s, the working shear stress in rivets,
f_B, the working bearing pressure on the rivets.

Single-riveted Lap Joint.

$$\text{Strength of plate} = (p - d)tf_t \text{.} \ldots\ldots\ldots\ldots (1)$$

$$\text{Strength of rivet} = \frac{\pi d^2}{4} f_s \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

$$\text{Bearing pressure on rivets} = dtf_B = \frac{\pi d^2}{4} f_s \text{.} \ldots\ldots\ldots (3)$$

If the plate strength is equal to the rivet strength, then, equating (1) and (3), p is found in terms of d.

$$(p - d)tf_t = dtf_B;$$

$$\therefore p = d\left(1 + \frac{f_B}{f_t}\right) \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

Also equating (2) and (3),

$$\frac{\pi d^2}{4} f_s = dtf_B;$$

$$\therefore t = \frac{\pi d}{4}\frac{f_s}{f_B} \text{.} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (5)$$

Allowing 6, 4, and 7 tons per square inch to be the respective values of f_t, f_s, and f_B as adopted in practice, and substituting these values in equations (4) and (5), then

$$p = 2{\cdot}16d \text{ and } t = 0{\cdot}45d.$$

These figures agree very well with general boiler practice, common proportions for a lap joint being

$$p = 2{\cdot}5d, \quad t = 0{\cdot}5d,$$

and the distance of the edge of the plate from the centre of the rivet is taken to be $1{\cdot}5d$.

For the general case of a lap joint, expressions (1), (2), and (3) become $(p - d)tf_t$, $\frac{n\pi d^2}{4}f_s$, and $ndtf_B$ respectively, where n is equal to 2 for double-riveted lap joints and equal to 3 for treble riveting. The corresponding value of p is therefore

$$p = d\left(1 + \frac{nf_B}{f_t}\right),$$

t remaining the same in all cases.

In general boiler practice the actual values of p and t for double riveting are $p = 3{\cdot}3d$ and $t = 0{\cdot}582d$; and for treble riveting $p = 3{\cdot}67d$ and $t = 0{\cdot}72d$. It will be noticed that the thickness of the plate in each of the three types of riveting in boiler practice is increasing compared with the value $0{\cdot}45d$ found theoretically. That is, in actual practice, smaller rivets and more of them are used than is given by the theory. In the theoretical treatment no account was taken of steam-tightness, and of course it does not at all follow that a joint of satisfactory proportions for withstanding a tensile test in a testing machine will be steam-tight. Probably the theoretical double- and treble-riveted lap joints would not be steam-tight, hence the necessity for making the actual rivet pitch smaller than that given by the theory.

Butt Joint with Single Cover Plate.—The strength of this joint is the same as that for a similar riveted lap joint. The strap must be at least as thick as the plate, and is better one and an eighth times as thick. Joints of this type are not used for boiler work.

Single-riveted Butt Joints with Double Butt Straps.

Strength of the plate $= (p - d)tf_t$ ....(1)

Strength of the rivet $= \frac{2\pi}{4}d^2f_s$..................(2)

Bearing pressure on rivet $= dtf_B$................(3)

As before, these three expressions are all equal, therefore,

equating (1) and (3), $p = d\left(1 + \frac{f_B}{f_t}\right)$, ...(4)

and (2) and (3), $t = \frac{\pi d}{4}\,\frac{f_s}{f_B}$. .... (5)

Substituting the same values for f_t, f_s, and f_B as in lap joints, then

$$p = 2{\cdot}16d, \text{ and } t = 0{\cdot}9d.$$

Double-riveted Butt Joints with Double Cover Plates.

Strength of plate $= (p - d)tf_t$.................(1)

Strength of rivet $= \frac{4\pi}{4}d^2f_s$.....................(2)

Bearing pressure on rivet $= 2dtf_B$..............(3)

From the equality of these expressions

$$p = d\left(1 + \frac{2f_B}{f_t}\right), \qquad (4)$$

and

$$t = \frac{\pi d}{2}\frac{f_s}{f_B} \qquad (5)$$

Substituting as before for f_t, f_s, and f_B, then

$$p = 3 \cdot 3d, \text{ and } t = 0 \cdot 9d.$$

Proportions from actual boiler work are

$$p = 4d, \text{ and } t = 0 \cdot 88d.$$

Treble-riveted Butt Joint with Double Cover Plates.

$$\text{Strength of plate} = (p - d)tf_t \qquad (1)$$

$$\text{Strength of rivet} = \frac{6\pi}{4}d^2 f_s \qquad (2)$$

$$\text{Bearing pressure on rivet} = 3dtf_B \qquad (3)$$

From the equality of the above expressions p and t are found to be, on substituting the same values as before,

$$p = d\left(1 + \frac{3f_B}{f_t}\right) = 4 \cdot 5d,$$

and

$$t = \frac{\pi d}{2}\frac{f_s}{f_B} = 0 \cdot 9d.$$

Again, steam-tightness has not been considered in obtaining these proportions, but they happen to give a steam-tight joint in this case and are fairly closely followed in practice, namely:

$$p = 4 \cdot 78d, \text{ and } t = 0 \cdot 89d.$$

A comparison of the results obtained from the foregoing theory with actual practice as found in boiler work, can be made by reference to the following table:

	Single Riveted.	Double Riveted.	Treble Riveted.
Lap joint—			
Proportions according to theory	$d = 2 \cdot 2t$ $p = 4 \cdot 75t$	$d = 2 \cdot 2t$ $p = 7 \cdot 26t$	$d = 2 \cdot 2t$ $p = 9 \cdot 9t$
Proportions according to boiler practice	$d = 2t$ $p = 4 \cdot 5t$	$d = 1 \cdot 8t$ $p = 5 \cdot 76t$	$d = 1 \cdot 1t + \frac{1}{4}''$ $p = 5 \cdot 1t$
Butt joint—			
Proportions according to theory	$d = 1 \cdot 11t$ $p = 2 \cdot 4t$	$d = 1 \cdot 11$ $p = 3 \cdot 66t$	$d = 1 \cdot 11t$ $p = 5t$
Proportions according to boiler practice	$d =$ $p =$	$d = 1 \cdot 13t$ $p = 4 \cdot 52t$	$d = 1 \cdot 13t$ $p = 5 \cdot 4t$

Efficiency of Riveted Joints.

$$\frac{\text{strength of joint}}{\text{strength of solid plate}} = \text{efficiency}$$

The strength of the joint is, of course, either equal to the strength of the plate where it is cut away by the rivet holes, or to the strength of the rivets, whichever is the smaller.

In the cases considered these are equal, so the strength of the joint may be taken to be proportional to $(p - d)$, when the strength of the unweakened plate is proportional to p. The efficiency of the joints considered is, therefore, in every case $\frac{p - d}{p}$.

From this expression the efficiency of a single-riveted lap joint is $\frac{2\cdot16d - d}{2\cdot16d} = 0\cdot54$, and for a treble-riveted lap joint the efficiency is $\frac{4\cdot5d - d}{4\cdot5d} = 0\cdot78$.

The rivet efficiency is calculated in the same way from:

$$\text{rivet efficiency} = \frac{\text{strength of rivet}}{\text{strength of solid plate}}.$$

Draughtsman's Method of Designing a Riveted Joint.—The type of problem put before a draughtsman may be: Design a treble-riveted butt joint with double cover plates, five rivets per pitch, for a boiler 16 ft. 3 in. diameter, steam pressure 155 lb. per square inch. The ultimate strength of plate material is 28 tons per square inch, and the factor of safety 4·5. The Board of Trade Regulation is that f_s is to be equal to $\frac{23}{28}f_t$, and double shear to be one and seven-eighth times single shear. Calculate the diameter of the rivet from some such rule as $d = t + \frac{1}{16}$ in., and t from $t = \frac{PD}{2f_t\eta}$, where P represents the boiler pressure, D the diameter of the boiler, and η the efficiency of the joint. The reason for adopting the rule $d = t + \frac{1}{16}$ in. is to ensure having a steam-tight joint. If the rivet diameter is less than t, although it might withstand the shearing force between the plates, it might not bring them sufficiently close together to give a steam-tight joint.

The draughtsman knows the probable strength of the joint, which in this case is approximately 85 per cent that of the solid plate (see p. 113).

From the data given,

$$f_t = \frac{28}{4\cdot5} = 6\cdot23 \text{ tons per square inch;}$$

$$f_s = \frac{23}{28}f_t = \frac{23}{28} \times 6\cdot23 = 5\cdot12 \text{ tons per square inch;}$$

$$\text{and} \quad t = \frac{PD}{2f_t\eta} = \frac{155 \times 195}{2 \times 6\cdot23 \times 0\cdot85 \times 2240} = 1\tfrac{5}{16} \text{ in.}$$

$$\text{Diameter of rivet} = t + \tfrac{1}{16} \text{ in.} = 1\tfrac{3}{8} \text{ in.}$$

The draughtsman would now ascertain the maximum pitch allowed by the Board of Trade, who say:

$$\text{maximum pitch} = (\text{C} \times \text{thickness of plate}) + 1\tfrac{5}{8},$$

where C has the following values:

Number of Rivets per Pitch.	(Lap Joint) C.	(Butt Joint) C.
1	1·31	1·75
2	2·62	3·50
3	3·47	4·63
4	4·14	5·52
5	—	6·00

The pitch must not exceed 10 in., however thick the plates. The maximum pitch allowed by the Board of Trade in foregoing problem is, therefore, $(6 \times t) + 1\frac{5}{8}$, since there are five rivets per pitch,

$$\text{i.e. } p = 6 \times 1\tfrac{5}{16} + 1\tfrac{5}{8}$$
$$= 9\tfrac{1}{2} \text{ in., say.}$$

The joint must now be checked, and to do this the actual plate efficiency must be calculated and compared with the assumed efficiency. If these two results differ very much, then the initial assumptions made must be revised.

$$\text{Actual plate efficiency} = \frac{p - d}{p},$$

$$\text{i.e. } \eta = \frac{9{\cdot}5 - 1{\cdot}375}{9{\cdot}5} = 0{\cdot}855.$$

The rivet efficiency must also be calculated and should be equal to or greater than the plate efficiency.

$$\text{Rivet efficiency (for this joint)} = \frac{5 \times 1\frac{7}{8} \times \frac{\pi d^2}{4} \times f_s}{p \times t \times f_t}$$

$$= 0{\cdot}92.$$

The actual pressure the boiler will carry may now be calculated by substitution in the formula,

$$\text{P} = \frac{2 f_t \, \eta t}{\text{D}},$$

$$\text{i.e. P} = \frac{2 \times 6{\cdot}23 \times 0{\cdot}855 \times 1\frac{5}{16}}{195}$$
$$= 162 \text{ lb. per square inch.}$$

Referring to fig. 14*b*,

$$V = \sqrt{(\tfrac{11}{20}p + d)(\tfrac{1}{20}p + d)}$$
$$= 3\tfrac{1}{2} \text{ in.};$$
$$\text{and } V_1 = \tfrac{1}{20}\sqrt{(11p + 8d)(p + 8d)}$$
$$= 2\tfrac{1}{2}.$$

The distance to the outside edge of plate from centre of rivet is usually taken as $1{\cdot}5d$, i.e. $2\tfrac{5}{8}$ in.

The thickness of the butt strap is calculated from

$$t_1 = \tfrac{5}{8}\left(\frac{p - d}{p - nd}\right)t$$
$$= \tfrac{5}{8}\left(\frac{9{\cdot}5 - 1{\cdot}375}{9{\cdot}5 - 2{\cdot}75}\right)1\tfrac{5}{16}$$
$$= \tfrac{5}{8} \times \frac{8{\cdot}125}{6{\cdot}75} \times \frac{21}{16}$$
$$= 1 \text{ in., say.}$$

Method of Stress Bands for Designing Riveted Joints.—The method of stress bands is due to Schwedler, and is sometimes used to check the calculations already made by any of the previous methods. Schwedler's method can perhaps be best explained by considering the foregoing example. The joint is a treble-riveted butt joint and the pitch of the rivets $9\tfrac{1}{2}$ in. The distance $(p-d)$, in this case $8\tfrac{3}{16}$ in., is divided into 10 equal bands, two bands to each rivet in the pitch. A circle is now drawn, round each rivet, of radius equal to the radius of the rivet plus width of one band, and curves drawn from these circles as in fig. 24 (facing p. 115). The bands so formed are supposed to represent the stresses in the plates and their direction of "flow". If the joint has been properly designed these stress bands illustrate each rivet having a portion of plate of the same strength, and any crowding together or spacing out of the bands clearly indicates at once the weakness of the joint, which can be rectified by altering mistakes made in the calculations or initial assumptions. The stress bands for the butt straps are also indicated in fig. 24, in which case it is readily seen that the bands round the rivets form continuous links similar to that of a chain. These bands are, however, less than the width of the bands for the plates, and illustrate the need to make the butt straps thicker than the theoretical thickness

Fullering.—Notwithstanding the great care with which a boiler may be built and riveted, the riveted plates gape at the edges and leak when under pressure test. To overcome this the edges of the plates are forced down on each other to form a close metal-to-metal joint by means of a fullering tool. This is done both inside and outside the boiler. The fullering tool, fig. 25 (facing p. 115), is shaped and used somewhat like a chisel. It has a very blunt square-shaped nose the same thickness as the plate on which it is used, and about $\tfrac{1}{8}$ in. bevel on the under side to prevent

any indentation or damaging of the inside plate. The burr which is left by the bevel is then fullered up by a second fuller without any bevel, thus leaving a good well-finished face on the edge of the plate as well as a fluid-tight joint. To facilitate fullering, all boiler plates should be machined to a taper of 1 in 8. Fullering must be done with great care and discretion, as too much, too heavy, or irregular application of the tool only tends to separate the joint again.

Rivet heads are treated in the same manner, and the same care should be exercised with them as with the plates.

APPLIED HEAT

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The authors desire to acknowledge the use they have made, in the earlier chapters of this section, of the late Dr. Draper's book, "Heat and the Principles of Thermodynamics", which has been placed at their disposal by the publishers, Messrs. Blackie & Son, Limited. Many of the subjects, for which there is only sufficient space to give a very short sketch here, are discussed in much fuller detail in that work.

Applied Heat

CHAPTER I

Introductory

Everyone is familiar with the sensations of "hot" and "cold". The physical cause of these sensations is termed *heat*, and the study of the physical phenomena accompanying these sensations is the study of heat.

If we mix some hot and some cold water together, the resulting mixture is cooler than the original hot, and hotter than the original cold water. "Something", which we will provisionally call "heat", appears to have left the hot water, leaving it cooler, and entered the cold water, making it hotter. Heat has been transferred from the hotter mass of water to the colder mass. *Heat, then, appears to be transferable, and to "flow" from hot bodies to cooler ones.*

Suppose we hammer a piece of cold iron on an anvil with a sledge-hammer. If we touch it, we shall find that it is much hotter after the hammering than it was before. It must have gained "something"—heat—but the hammer does not appear to have *lost* "anything"; on the contrary, it is slightly hotter. But the experimenter has lost something if he has done the experiment vigorously, and is possibly gently perspiring from the exertion of the exercise. He has expended a considerable amount of what is called **mechanical energy** and must rest to recuperate himself. The "something" that the iron possesses that it did not before is an *extra* amount of *energy*. Heat appears to be a form of energy, and, if it is, it should be measurable, just as other forms of energy are. We may suppose, meantime, that *heat is a measurable "something", namely ordinary mechanical energy. The process of measuring quantities of heat is called* **calorimetry.**

Kinetic Theory of Heat.—In the early years of the nineteenth century heat was considered to be a kind of fluid—*caloric*, as it was called. It was one of those fairy fluids which have diffused themselves through physical theory—old and new. It was imponderable; could penetrate

solids; could pass from one body to another. Count Rumford, Sir Humphry Davy, and Joule finally disposed of it, and substituted the kinetic theory of heat, which, at present, seems to be fairly well established. According to this theory, heat is regarded as molecular motion, and heat energy as the energy of molecular motion. When we "heat" a body, we make its molecules move about faster. One of the most characteristic changes that accompany the increased motion of the molecules is an increase of *temperature*.

Temperature.—If we have 11 bodies, say, we can arrange them in their correct order of hotness by feeling them, if each is sufficiently different in "hotness" from the others.

The scheme might be this:

A	B	C	D	E	F	G	H	I	J	K
0	1	2	3	4	5	6	7	8	9	10

A is the coldest body, K the hottest, and the body D, say, is hotter than C and colder than E.

We can ascribe serial whole numbers to the bodies, calling A, 0, and K, 10. The number which any intermediate body must have is then definite, say 7.

We could speak of the hotness of D as being of **degree** 3. Instead of "degree of hotness" we use the word **temperature,** and instead of relying on our sense of hot and cold to put bodies in their proper order of hotness, we use a **thermometer.** In this instrument the expansion, by heat, of a liquid in a closed glass tube can be watched and measured. The hotter the body, the greater is the expansion of the mercury in the thermometer.

In settling the **scale of temperature,** we choose melting ice for the body A and call its temperature 0, and boiling water for the body K and call its temperature 100. The *distance* between the marks on the thermometer stem, corresponding to these temperatures, is divided into 100 equal parts, and each division is given a serial number,

$$0^\circ \text{ C.},\ 1^\circ \text{ C.},\ 2^\circ \text{ C.},\ 3^\circ \text{ C.},\ \ldots\ 100^\circ \text{ C.},$$

where ° C. stands for *degrees centigrade.*

The scale is extended for temperatures above 100° C. and below 0° C. by extrapolation. Mechanical engineers use the Fahrenheit thermometer more than the Centigrade one, and this thermometer will be used henceforth. It is described on p. 139.

Mechanical Equivalent of Heat.—If our supposition that heat is molecular kinetic energy be true, then equal amounts of mechanical energy should correspond to equal amounts of heat, if all the energy is transformable. This deduction can be easily tested experimentally by using an electric current to supply energy to a given mass of water.

Suppose R is the resistance of a wire in ohms, and E the electrical pressure across the wire in volts, then the current flowing is

$$\frac{E}{R} \text{ amperes;}$$

and this current flows under the electrical pressure of E volts, hence the rate of working of the current in the resistance is

$$\frac{E^2}{R} \text{ watts.}$$

Now it is shown in books on electricity that 1 h.p. is equivalent to 746 watts.

$\therefore$ 550 ft.-lb. per second is equivalent to 746 watts,

$\frac{550}{746}$ " " " 1 watt,

i.e. 0·737 ft.-lb. per second is equivalent to 1 watt, hence the mechanical energy developed in the wire in t sec. is

$$0{\cdot}737\,\frac{E^2}{R}t \text{ ft.-lb. or } 0{\cdot}737\,C^2Rt \text{ ft.-lb.}$$

The arrangement of the circuit is indicated in fig. 1, a source of current, an ammeter, and the wire wherein the heating effect is to be observed being placed in series. Fig. 2 shows a common laboratory form of calorimeter. The wire A is fastened to thick terminals, which are connected with the binding screws d, d, which are joined up to the circuit wires. Through the wooden lid of the calorimeter also pass a thermometer b and the rod of a stirrer e. In Joule's apparatus the wire A was of platinum-silver, in length about 4 yd., bent back on itself like the coils of a resistance box, and fastened to a glass tube, the whole being coated with shellac varnish. Its resistance was 0.9895 ohm. The vessel F was of copper, and contained about a gallon of distilled water. It was supported on a wooden frame and carefully protected from draughts, and to diminish radiation it was surrounded with a layer of silk.

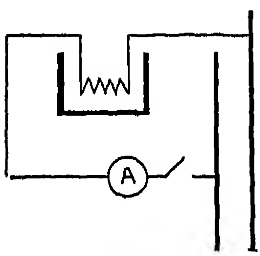

Fig. 1

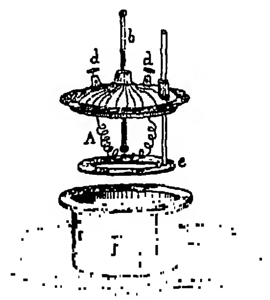

Fig. 2

The following experiments may be carried out with this apparatus.

Put about 1 qt. of water in the calorimeter, taking care to cover the wire with water, and pass sufficient current through the wire to give a rise of

temperature of about 40° F. in 5 min. Repeat this experiment with different amounts of water, keeping everything else the same. Take 2 qt., 3 qt., and 4 qt. Tabulate the figures thus:

Amount of Water in Pounds.	Rise in Temperature.	Product of Mass × Rise of Temperature.
2·50	40° F.	100
5·00	20° F.	100
7·50	13·3° F.	100
10·00	10° F.	100

The figures will come out round about those given in the table.

Now, ignoring secondary corrections, the same amount of energy was put into the water ($\cdot 737\ C^2Rt$) in each of these experiments, since the resistance and current in the wire were not changed, nor was the time that the current was flowing altered.

Consequently, if heat is a form of energy, the same amount of heat must have been given to the water in each experiment. The rises of temperature are different, *hence temperature cannot measure heat*, but the product of the mass of water into the rise of temperature is constant.

Hence if *we measure heat by the product of mass into rise of temperature* we have a measure which is not inconsistent with the supposition that heat is a form of energy. This experiment leads at once to a definition of a unit of heat.

Definition of a Unit of Heat.—A unit of heat is *defined* as *the amount of heat required to raise unit mass of water through 1 degree of temperature.* A precise definition should state at what part of the temperature scale the change of temperature is to be measured. Using the pound as the unit of mass and the Fahrenheit scale of temperature, **the British Thermal Unit** (written B.Th.U.) is practically defined as *the quantity of heat required to raise 1 lb. of water 1 degree F.*

Having now determined upon a unit of heat, this unit of heat must be equivalent to so much mechanical energy. If J is the number of foot-pounds of energy in one unit of heat, i.e. in 1 B.Th.U., then

$$0{\cdot}737\ C^2Rt = JH,$$

where H is the heat developed in B.Th.U. in time t, i.e.

$$\frac{0{\cdot}737\ C^2Rt}{H} = J.$$

J is called *the mechanical equivalent of heat* and can be easily found approximately with the electrical calorimeter. In one experiment the current was 3·073 amperes, the resistance of the wire 0·9895 ohms. E was therefore 3·04 volts. The current was on for one hour.

The heat required to raise the temperature of the water and the appa-

ratus 1° F. was 13·4 B.Th.U., and the temperature actually rose by 2·36° F., whence

$$H = 2{\cdot}36 \times 13{\cdot}4,$$
$$J = \frac{0{\cdot}737 \times 3{\cdot}073^2 \times 0{\cdot}9895 \times 3600}{2{\cdot}36 \times 13{\cdot}4}$$
$$= 781 \text{ ft.-lb. per B.Th.U.}$$

The more accurate value, when every refinement is allowed for in the measurements, is 778 ft.-lb. per B.Th.U.*

Effects of Heat.—The effects which heat produces are very various; nearly every property of a body is affected by the addition or withdrawal of heat. The principal effects to be here treated of are: (i) change of temperature, (ii) change of volume, and (iii) change of physical state.

(i) *Change of Temperature.*—We have already referred to this effect of heat.

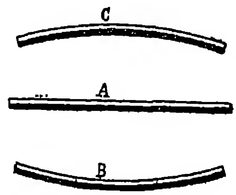

Fig. 3

(ii) *Change of Volume.*—If a small quantity of air be tied up in a bladder and placed near a fire, a large increase of size is at once apparent. If a flask exactly full of water be heated, some of the water will overflow. If an iron wire be tightly stretched between two supports and then heated strongly, its length will be visibly increased.

These and numerous facts of a similar nature establish the general rule, that *bodies expand when heated and contract when cooled.*

If a very hot liquid be poured into a glass vessel, the vessel is liable to be cracked, owing to the strains set up by the expansion of the inner portion while the outer surface is still cold. Glass touched with a piece of hot metal will crack.

The riveting together of boiler plates is generally done with red-hot rivets, which on cooling contract and pull the plates very close together. The same principle is taken advantage of in the building of heavy guns. These are formed of a number of concentric cylinders of steel which are successively slipped into position in a red-hot condition, the inner cylinder or cylinders being cold. The enormous pressure upon the innermost cylinder, which results from the contraction of the outer ones, enables it to withstand the pressure of the gases produced by the explosion of heavy charges of gunpowder. A similar process is adopted in fitting iron tyres on wheels. Railway metals are laid with a small space between them to allow for contraction and expansion. Furnace-bars for like purpose are never tightly fixed, and gas-pipes are laid with telescopic joints.

The amount of expansion for any given rise of temperature is different for each substance. This may be illustrated by riveting together thin bars of different metals. Then, if these compound bars are straight at ordinary temperatures, they become curved when they are either heated or cooled. Thus, if in the bars shown in fig. 3 the unshaded portion represent silver

* The scientific value is $4{\cdot}185 \times 10^7$ ergs (Barnes, 1909).

and the shaded portion iron, on a rise of temperature the bar bends as in C; on a fall of temperature it assumes the form shown in B, since silver elongates or contracts more than iron for the same change of temperature.

(iii) *Change of Physical State.*—It is a fact of common experience that some substances, e.g. water, can exist in more than one state, as a solid, a liquid, or a gas; and that under ordinary atmospheric pressure the state of the substance is determined by its temperature. The liquid is the intermediate stage; addition of sufficient heat converts a liquid into a gas; subtraction of sufficient heat converts a liquid into a solid.

Fusion or **melting** is the name given to the process of change that takes place when a solid becomes a liquid; its converse is **solidification.**

When a liquid passes into the gaseous state the process is called **vaporization;** and the converse process, when a substance passes from the gaseous to the liquid state, is **condensation.** Vaporization may take place in two ways, by **evaporation** and by **ebullition** or **boiling.**

Some solids, such as arsenic and camphor, readily pass from solid to gas or from gas to solid without becoming liquid at all; these are said to sublime, and either process is called **sublimation.**

If a mass of pounded ice at freezing-point be placed in a vessel and a little hot water be poured upon it, a thermometer immersed in the ice indicates no rise of temperature; whereas, if the experiment be repeated with a mass of water at freezing-point, a rise of temperature results. And again, water under ordinary conditions boils at a certain temperature called the boiling-point, and cannot be made any hotter however high the temperature to which it may be exposed.

When heat is added to a substance, without rise of temperature, the heat is said to be rendered **latent.** Latent heat is heat that is absorbed by the substance during the change of state from solid to liquid, or from liquid to gas. It does not raise the temperature of the substance.

Sources of Heat.—The prime source of our heat is, of course, the sun. By his rays the general temperature of the surface of the earth and of the atmosphere is maintained. The heat reaching us directly from the sun is also made use of occasionally to accomplish definite objects, e.g. the extraction of salt from sea water.

For special purposes, however, we draw upon secondary sources. Heat may be produced by chemical action, by change of physical state, and by currents of electricity. It is also produced in mechanical operations such as friction, collision, compression, torsion, &c., during change of the magnetic condition of magnetized bodies, and by spontaneous changes taking place in the structure of the atoms of certain bodies.

(i) *Chemical Action.*—The ordinary process of **combustion** of fuel is a chemical combination, usually between carbon and oxygen or between oxygen and hydrogen. The heat of our bodies is maintained by exactly similar processes, which take place at a slower rate and thus produce only a moderate temperature.

(ii) *Change of Physical State.*—If steam be passed into water, the steam condenses and gives out large quantities of heat during the change. This process takes place on a large scale in the atmosphere, and the heat given out during the condensation of the water vapour into rain has an important effect on climate. And similarly in solidification: if a beaker of water at 32° F. is placed in a freezing mixture, the temperature remains zero while freezing of the water proceeds.

(iii) *Currents of Electricity.*—If an electrical current be passed through a wire, the wire becomes more or less heated. This is a convenient method of applying heat in particular cases.

(iv) *Friction.*—It is well known that a small piece of iron may be made red-hot by rapid blows with a hammer; a leaden bullet on striking a target is often melted; heat and sparks are produced when a railway train is stopped by a brake; and when timber is sawn or metal filed great heat is produced.

(v) *Radio-activity.*—The recent discovery of the radio-active substances indicates that there is probably a store of latent energy within the atoms of some kinds of matter which is gradually assuming the form of heat. The emission of heat from a gramme of radium amounts to as much as 0·470 B.Th.U. per hour, and the process continues for many hundred years. Estimates of the amount of radio-active material present in the earth, and of the rate of its heat emission, indicate that the heat from this source may be an important factor in the maintenance of the heat of the earth and perhaps of the sun. The heat is supposed to be derived from the internal energy of the radium atom. The atoms of elements are supposed, at present, to be systems of small electrified particles called protons and electrons. Each atom possesses a store of energy, some of which it gives up when the constitution of the atom changes.

Modes of Transference of Heat.—Heat may be transferred from one body to another by **conduction**, by **convection**, and by **radiation.**

(i) *Conduction.*—If a short metal rod be held with one end in a flame, the other end becomes warm. The heat has passed from layer to layer of the rod, passing from the portion which is at a higher temperature to the portion which is at a lower temperature. This is the distinguishing feature of conduction.

(ii) *Convection.*—When hot water is admitted into one end of a bath and cold water into the other, there may be great difference of temperature between the ends of the bath. If the water be stirred, the whole takes the same temperature. The heated water has travelled from one part to another, carrying its heat with it. This process is called convection. It is the principal mode by which heat is distributed through the mass of a fluid. Owing to the expansion that takes place on heating, the density of a fluid that is changing temperature is generally different in different parts. If the liquid be heated at the bottom, the force of gravity at once causes the upper, i.e. the denser portions, to fall and the less dense lower portions to rise, currents of fluid being thus set up which tend to equalize

the temperature throughout. If a vessel of liquid be heated at the top these currents are not produced.

Winds are produced by the unequal heating of the air in different parts of the world by the sun's rays.

Hot-water or hot-air pipes are often used as channels by which a hot fluid may pass through and warm a building. The distinguishing feature of convection is that masses of a fluid serve as vehicles to convey the heat from place to place.

(iii) *Radiation.*—Heat often passes from one body to another without conduction or convection. If the hand be held in front of a fire the warmth is instantly felt, although a draught of air may be proceeding from the hand to the fire. And, moreover, heat passes with great facility through an ordinary vacuum and through some substances without warming them, conditions completely at variance with the processes of conduction and convection. This mode of transference is radiation. Its distinguishing features lie in the fact that the temperature of the medium may be unaffected, and the presence of a "material" medium is not essential.

Nature of Heat.—The caloric theory failed to give any reasonable explanation of the production of heat by friction, in which process *both* bodies, the rubbed and the rubbing, become *hotter*.

On the hypothesis that heat was a material fluid, where did the heat thus manifested come from? Two suggestions were made: first, that it came from surrounding bodies; and secondly, that the thermal capacities of the two bodies rubbed together were diminished in the process, so that the quantity of caloric actually present was sufficient, under these conditions, to make the bodies hot. About 1795 Count Rumford, while engaged in boring cannon at Munich, turned his attention to the theory. His cannon became so hot that when surrounded by water they caused the water to boil. The caloric did not then come from the water surrounding the cannon, for this was hotter than before. Neither was the thermal capacity of the borings diminished by the disintegration of the material of which they were the parts. The suggested explanation being thus inadequate, Rumford concluded that heat was not a material fluid, seeing that it could be produced to any amount, that the then-received explanation of the heat produced by friction was not tenable, and that it was difficult to conceive anything to have been communicated to the mass of metal he operated upon except it were *motion* or a property of motion.

About the same time Sir H. Davy melted two pieces of ice by rubbing them together. It was known that the water produced had not a smaller but a greater thermal capacity than the ice, and the second of the suggestions mentioned above became untenable.

These experiments showed the caloric theory to be quite incapable of explaining the production of heat by friction. The hypothesis that the phenomena of heat are due to the motion of the molecules of which all substances are composed, has now been apparently established.

CHAPTER II

Temperature

Methods of Estimating Temperature.—Changes of temperature usually change the physical properties of bodies; for instance, iron at a high temperature is liquid; at a low temperature, solid. Some of these changes can be conveniently used to measure temperature. The following modes of measuring temperature are in use.

(i) By measuring change of volume. The liquids most commonly used are mercury, alcohol, pentane, or toluene. They are contained within a glass or porcelain reservoir. The rough indications afforded by solids are usually based on change of length.

(ii) By measuring change of pressure of gases.

(iii) By calorimetric measurements. When a portion of some substance is taken from a place of high temperature and immersed in water, the temperature of the water rises. The rise of temperature may be used to find the temperature of the hot body.

(iv) By measuring the change of an electrical property, such as the thermo-electric force of a "junction", or the resistance of a wire, as in the case of Langley's bolometer and the platinum thermometer.

(v) By measuring the radiation emitted by a hot body.

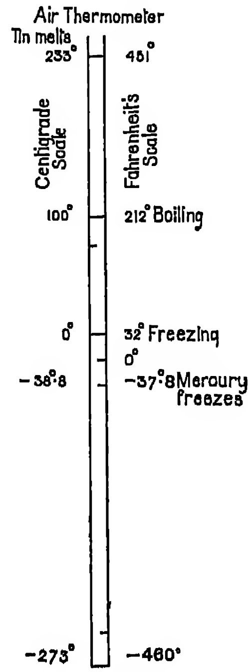

Fig. 4

It should be borne in mind that, at present, we regard temperature as merely a *scale*. A body at a temperature called 60 would *feel* much hotter than a body at 6. It is desirable to think very carefully over the step from the "sense-impression" to the instrument. In everyday engineering practice, we measure temperatures so frequently with thermometers that we are apt to forget the basis upon which the whole thing rests. The line of argument is this:

1. Suppose we have ten bodies A, B, C, . . . J. We can set them in their order of "hotness" with our fingers, if there is sufficient difference in hotness between any pair.

2. We ascertain by experiment that the volume of a given mass of gas *at constant pressure* varies when it is at the different degrees of hotness corresponding to those of A, or B, or C, &c., and that the higher the degree of hotness, the larger is the volume.

3. If then we arranged the bodies A, B, &c., in accordance with the volume of a given mass of gas at constant pressure, in contact with them, the series would be in the same order as in 1.

4. Consistently with 1 then we can measure "temperatures" in this way with the "constant-pressure gas thermometer".

5. We shall now be able to arrange bodies with a much finer gradation of temperature than was possible with our fingers only, and shall have every confidence that, were our senses sufficiently acute, the conclusions drawn from the "gas thermometer" would coincide with our sense-impressions.

It was by reasoning on such a basis that Galileo introduced the first thermometer—the *gas thermometer*. The gas thermometer is still the standard, but to-day it is the *constant-volume gas thermometer* that is the standard. In this thermometer the *pressure* changes with the temperature of the body.

Graduation of the Constant-pressure Air Thermometer.—We will now consider in its utmost simplicity the principle upon which a constant-pressure gas thermometer is graduated.

Take a tube of glass, say a metre long, of uniform bore of about 1 mm., and having one end sealed (fig. 4). Warm the tube, and then at once dip the open end of the tube into some coloured liquid—a little water with red ink in it does very well. Allow the tube to cool. As it cools it will suck a thread of liquid, say a centimetre long, into the tube. Remove the end of the tube from the coloured liquid, and as the tube cools further, the thread of coloured liquid will be sucked into the tube and will form an index which divides the air within the thermometer from the air outside. This instrument is, in principle, a constant-pressure air thermometer.

Get a jar and fill it with water into which a number of broken lumps of ice have been put. A mixture of ice and water in which ice is actually melting we shall take as one of our standard bodies, say body A. Put the thermometer into the cold water so that the index is just not below the water-level; the coloured index will take up a steady position which corresponds to the volume of the enclosed gas at atmospheric pressure and at a temperature corresponding to melting ice. Make a scratch on the tube at the inner end of the coloured index. Now remove the thermometer and place it in steam from boiling water. We will suppose that the barometer reads 30 in. of mercury on the day when we make the experiment. The hot steam rapidly makes the gas in the thermometer expand, and the index takes up a higher steady position. Again make a scratch at the inner end of the coloured index. We now have two scratches on the stem of our thermometer, one corresponding to the temperature of melting ice and the other corresponding to the temperature of steam at atmospheric pressure. Now divide the *distance* between these two marks into equal parts, 100 if we want a centigrade scale, and 180 if we want a Fahrenheit scale.

Scales of Temperature commonly used.

(1) *Centigrade Scale.*—In the centigrade scale the mark on the stem corresponding to the temperature of melting ice will be labelled 0, and the mark corresponding to the temperature of steam, 100. There will be 99 division marks between these two extreme marks, and the temperatures corresponding to any consecutive pair of divisions are said to differ by 1 degree centigrade (1° C.).

(2) *Fahrenheit.*—The Fahrenheit scale is used in this country and the United States of America by engineers. On this scale the mark on the stem corresponding to the temperature of melting ice will be labelled 32, and the mark corresponding to the temperature of steam will be labelled 212, and there will be 179 marks of division between these two points, and the difference in temperature between the two standard points will be 180 degrees Fahrenheit (180° F.).

Rules for Conversion of Temperature Scales.—The rules for conversion of temperature scales are now fairly obvious. In the Fahrenheit scale we have 180 divisions corresponding to 100 centigrade divisions, but the Fahrenheit scale begins at 32 instead of 0. If F is a reading of a certain temperature in degrees Fahrenheit, then (F −32) would be the reading if we took the zero of the Fahrenheit scale at the temperature of melting ice. If the gas thermometer were put into a liquid at temperature F, *the index would go to exactly the same place on the stem,* whether the stem were graduated in Fahrenheit degrees or centigrade degrees. Consequently

$$\frac{(F - 32)}{180} = \frac{C}{100},$$

$$\text{i.e. } F = 1{\cdot}8\, C + 32,$$

where C is the centigrade reading corresponding to (F) °F. This is the formula to change one scale into the other.

The Absolute Scale.—Suppose that, instead of marking the freezing-point of water 32° F., we mark it $(32 + \tau)^\circ$ F., then the normal boiling-point of water will be marked $(212 + \tau)^\circ$ F. We can choose τ so that the volume of the air is exactly proportional to the temperature. If the volume be proportional to the temperature, when measured on this new scale, we have

$$\frac{\tau + 212}{\tau + 32} = \frac{\text{volume of air at normal boiling-point}}{\text{volume of air at normal freezing-point}}.$$

The right-hand fraction can be observed. Careful experiments give, as the ratio for air, 1·3665, hence

$$\frac{\tau + 212}{\tau + 32} = 1{\cdot}3665,$$

$$\text{i.e. } \tau = 459,$$

so that we add 459 to the normal Fahrenheit reading in every case, i.e.

$$T = 459 + t,$$

where T is measured on the new scale and t on the ordinary Fahrenheit scale.

The new scale is called the *absolute scale of the constant-pressure air thermometer*.

Since $T \propto V$, it follows that $V = 0$ when $T = 0$, so that at the zero of this scale the air would entirely vanish. This could not happen physically—in other words, we cannot expect by any physical means ever to attain the absolute zero of temperature on this scale.

Charles' and Gay-Lussac's Law.—The physicists Charles and Gay-Lussac found that, if a thermometer is constructed as already described, it is *immaterial which of the permanent gases is used to fill it*. The so-called *permanent* gases are oxygen, nitrogen, air, hydrogen, helium, &c.

This is an important physical fact, viz. that if temperature is defined so that $T = f(p)V$ for one permanent gas, the same law holds for *all* permanent gases.

Later experiments showed that the law is not quite accurate. By comparison of the best data available, the number to be added to the Fahrenheit reading to get the absolute temperature is taken as 460 instead of 459, the figure for the air thermometer; so that

$$T = 460 + t$$

is the relation actually used in engineering.

Boyle's Law.—Boyle discovered another fact of nature. If P is the pressure of a definite mass of a given gas and V its volume, then, so long as its temperature T is kept constant,

$$PV = \text{constant, very nearly,} \quad \text{.............. (1)}$$

i.e. the pressure is inversely proportioned to the volume.

The law may therefore be put

$$PV = F(T), \quad \text{..................... (2)}$$

where F(T) is some function of the absolute temperature of the gas. If T is constant, F(T) is necessarily constant.

This law applies to all permanent gases, but the constant is different for different gases.

Equations (1) and (2) can be combined into one equation,

$$\frac{PV}{T} = R', \quad \text{..................... (3)}$$

where R' is a constant. This constant is different for different gases

If P is constant,
$$\frac{V}{T} = \frac{R'}{P} = \text{constant.}$$

$$\therefore \frac{V_t}{V_{t'}} = \frac{T}{T'}.$$

If T is constant,
$$PV = R'T,$$

$$\text{i.e. } PV = \text{constant—see equation (1).}$$

The Gas Equation.—The equation

$$\frac{PV}{T} = R',$$

$$\text{or } PV = R'T$$

is the famous *gas equation*. It holds *exactly* only for *perfect gases* (by definition of a "perfect" gas), but it holds very approximately for all permanent gases under usual conditions. It begins to break down at very low temperatures and very high pressures for actual gases, as the gases begin to become vapours and subject to condensation under these extreme conditions.

Value of R′.—If we deal with *unit mass* of a gas and write $R' = \frac{R}{m}$ where m is the molecular weight of the gas in question, the gas equation becomes

$$\frac{PV}{T} = \frac{R}{m}, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where R is a universal constant.

In this formula

P stands for the pressure;
V, the volume of *unit mass*;
T, the absolute temperature of the gas.

The value of R is given in the following table:

Quantity.	F.P.S. (used in Britain).	C.G.S. (used in Science).
P	Pounds per square *foot*	Dynes per square centimetre
V	Cubic feet	Cubic centimetres
T	$(460 + t° \text{ F.})$	$(273 + t° \text{ C.})$
m	Molecular weight	Molecular weight
R	1545	$83{\cdot}15 \times 10^6$
Unit mass	Pound	Gramme

Thermodynamic Temperature.—The measurement of temperature can be placed on an entirely different basis, using the principles of thermodynamics.

Anticipating one of the results of thermodynamics which are discussed at the end of this article, we may mention here that Carnot showed that *temperature* is the quality which determines the efficiency of ideal engines, i.e. given a hot body at one temperature and a cold one at a lower temperature, *all* perfect engines working between these two temperatures have the same efficiency, no matter what the cycle of operations or the nature of the working substance may be. The efficiency of ideal engines can therefore be used to *define* a system of measuring temperature, so that none but *mechanical* measurements are needed to arrive at *the temperature* of a body. We can thus banish completely from our minds *all* arbitrary thermometers whether

filled with gas, air, or mercury; even that familiar spectre, the thermometer filled with a " perfect " or " ideal " gas, need haunt us no longer. Lord Kelvin worked out the theory of this method of defining and measuring temperature (see article "Heat", 9th ed., *Encyclopædia Britannica*), and showed that the engineer actually uses a thermodynamic thermometer when he determines the temperature of a boiler by measuring the boiler pressure and looking up the temperature in a steam table—for he showed that the temperature (as he defines it) can be *calculated* from the observed *mechanical* properties of steam. After laying the foundation of his method and showing how an absolute vapour-pressure thermometer can be constructed and calibrated by calculation to read thermodynamic temperatures, he examined what properties a substance must have for the volume of a given mass of it at constant pressure to be proportional to its thermodynamic temperature, i.e. he examined the possibility of a thermometer, based on the principle of proportional expansion, reading thermodynamic temperature. He found that the permanent gases possess the necessary properties with a high degree of approximation, and devised the porous-plug experiment to test the point. The constant-pressure gas thermometer is thus shown to measure absolute temperatures, as he defines them, very closely indeed.

The " gas " absolute scale and the thermodynamic absolute scale are thus, for engineering purposes, identical.

If a constant-*volume* gas thermometer is used, we must, in addition, know accurately the characteristic equation, at constant temperature, for the gas used. For ordinary engineering purposes this equation is simply the equation expressing Boyle's law, viz.

$$PV = f(T).$$

Details of the work described in this section are given in Kelvin's article on " Heat ", *Ency. Brit.*, 9th ed., and in Kelvin, " Math. and Phys. Papers ", Vol. I, p. 333. The reader who is interested in this matter is strongly recommended to read the *Encyclopædia Britannica* article.

We can now answer a question that is often asked in engineering discussions—what do we really mean by saying that the temperature of a furnace is, say, 2000° F. absolute? We mean that if a perfect engine worked between this temperature and a temperature of 500° F. absolute, say, the efficiency of the engine would be

$$\frac{2000 - 500}{2000} = \cdot 75, \text{ i.e. } 75 \text{ per cent.}$$

This result is calculated from the formula given on p. 205, namely, the efficiency (η) of a perfect engine is given by

$$\eta = \frac{T_1 - T_2}{T_1},$$

where T_1 is the upper and T_2 the lower temperature on the thermodynamic scale, between which the engine is supposed to work.

Practical Thermometry.—So far we have constructed, in theory only, an air thermometer in which the air expands when heated at constant pressure, and the expansion of the air is used as a basis upon which a thermometric scale is constructed. We will now consider the practical side of the question.

The standard thermometer of the physical laboratory is the constant-*volume* hydrogen thermometer. This thermometer is chosen because it is far easier to maintain a small and constant volume of hydrogen at a uniform temperature than a large and changing one. In this instrument, it is the pressure that varies with temperature. The pressure rises by $\frac{1}{492}$ part for a change of temperature from 32° F. to 33° F. A great deal of research has been carried out with this thermometer, and many standard temperatures have been determined with it, but it is a very difficult instrument to use. The principle involved in it is simple enough. A small bulb of anything from 50 to 200 c. c. in capacity is filled with hydrogen and heated in a furnace to a uniform temperature, and by means of a mercury-pressure-gauge the volume of the gas is kept constant. Certain standard temperatures are measured—for example, the melting-point of nickel—by observing the increase in the pressure of the hydrogen in the bulb. A resistance-coil is next heated to the same temperature and its resistance carefully measured. Several temperatures between the limits covered in the series of experiments are examined in this way, and a curve obtained which connects the resistance of the coil with the hydrogen thermometer temperature. The resistance of the coil can then be used to determine intermediate temperatures, in terms of the constant-volume hydrogen thermometer. Readings have been taken of different standard temperatures, ranging from about $-434°$ F. to about $+2820°$ F.

At very high temperatures the difficulties of experiment become very great. Both the bulb and the capillary tube of the thermometer must be made of a highly refractory substance. In the most recent determinations a platinum-rhodium alloy was used and nitrogen was used instead of hydrogen. At the high temperatures used the gas, at constant volume, exerts considerable pressure, and if this pressure had to be sustained by the heated bulb, its deformation would completely vitiate the results. For this reason the bulb is contained in another vessel in which the pressure is so adjusted, at each temperature, that it exactly balances the pressure inside the bulb. The bulb itself is therefore called upon to sustain no resultant pressure. The next difficulty arises from the porosity of the bulb. At the very high temperatures used the alloy is near its melting-point, and is not gas-tight. It is less porous to nitrogen than to hydrogen, hence the use of nitrogen. The highest figure reached, with confidence, appears to be about 2820° F.

In the low temperature direction Henning carried out an investigation from 30° F. to $-330°$ F. The lowest temperatures were obtained by liquid-air baths.

The Corrections of the Gas Thermometer.—We have already stated that the standard practical gas thermometer is the constant-volume hydrogen thermometer, and have mentioned, in connection with high-temperature research, the use of nitrogen. The facts are as follows:

1. The theoretical standard scale of temperature is the thermodynamic one which has been briefly explained on p. 135.

2. The constant-pressure gas thermometer can be graduated so that the volume of the gas in it is proportional to the " temperature ". Such temperature readings are identical with thermodynamic temperature readings, provided the molecular potential energy of the gas used is negligible. Kelvin devised the porous-plug experiments to test the point. When the potential energy is not negligible, the porous-plug (and other) experiments supply the data for calculating the corrections to the gas thermometer readings.

3. If a constant-*volume* gas thermometer is used we have to compute what the reading of the corresponding constant-pressure gas thermometer would be. If the gas accurately obeys Boyle's Law, this is easy, for the product of the pressure and volume of any given mass of the gas is constant at a definite constant temperature. If the gas does not obey Boyle's Law exactly, we have to allow for this in our computation.

4. The corrections to be added to the gas scale readings to obtain thermodynamic temperatures are given in the following table:

CORRECTIONS TO THERMODYNAMIC SCALE $\theta_0 = 273{\cdot}10°$ C.

(From *Methods of Measuring Temperature*, by E. Griffiths, 1918, p. 15. Publishers: Griffin & Co.)

Temp. Centigrade.	Constant pressure = 76 cm.			Constant volume p_0 = 100 cm.		
	Helium.	Hydrogen.	Nitrogen.	Helium.	Hydrogen.	Nitrogen.
− 250	—	—	—	+0·02	—	—
− 200	+0·10	+0·26	—	+0·01	+0·06	—
− 100	+0·03	+0·03	+0·33	0·000	+0·014	+0·07
− 50	+0·009	+0·004	+0·09	0·000	+0·004	+0·02
+ 25	−0·002	−0·002	−0·013	0·000	0·000	−0·006
+ 50	−0·002	−0·003	−0·017	0·000	0·000	−0·006
+ 75	−0·002	−0·002	−0·012	0·000	0·000	−0·004
+ 150	+0·005	+0·003	+0·04	0·000	+0·001	+0·01
+ 200	+0·01	+0·01	+0·10	0·000	+0·002	+0·04
+ 450	+0·07	+0·04	+0·50	0·00	+0·01	+0·15
+1000	+0·24	—	+1·7	—	+0·04	+0·70
+1500	—	—	+3·0	—	—	+1·3

The comparatively recently discovered atmospheric gases, helium, argon, neon, &c., are monatomic, and the table shows how very small the helium corrections are. These gases possess several very valuable properties for high-class thermometric work; they are chemically inert and have, especially helium, a very low point of liquefaction.

Mercury Thermometer.—Although the gas thermometer which we have so far described is a practical standard thermometer, it is quite unsuitable for ordinary practical purposes. A large number of standard

temperatures have been observed carefully by comparison with the gas thermometer, and these standard temperatures are always available and are easily reproduced, and can therefore be used to calibrate simpler thermometers. The simplest of these secondary instruments is the well-known mercury thermometer.

We need not spend time in describing the appearance of this instrument. For engineering purposes the instrument is graduated in degrees Fahrenheit.

Details of the Mercury Thermometer.—Mercury freezes under atmospheric pressure at $-38°$ F. and boils at $675°$ F. It can be used to measure temperatures up to about $900°$ F., but temperatures which are above the boiling-point of mercury can only be reached by special methods. The usual method is to fill the tube above the mercury with nitrogen at a sufficiently high pressure to prevent the mercury from boiling at temperatures less than $900°$ F. A hard glass, usually a boro-silicate glass, is used. In engineering practice the thermometers are usually sent to a physical laboratory to be calibrated. In using the instrument it must be remembered that the calibration is made with *the whole* of the mercury column immersed in the bath, the temperature of which is being taken.

"Exposed Thread" Correction.—In practice it is rarely possible to immerse the whole of the thread of mercury. For instance, if we are taking the temperature of superheated steam by putting the thermometer into a thermometer pocket in the steam-pipe, probably an inch of the thread sticks out of the pocket. In such cases a correction must be made. The reason for the correction is this: in calibrating the instrument the whole thread was immersed at the temperature read. Consequently it was subject to the full "expansion-in-glass" corresponding to the temperature. If the entire thread is not subjected to the full temperature the expansion is not so much, and consequently the thermometer appears to read lower than it really should. The correct reading will therefore be higher than the apparent reading. The correction is made as follows. An auxiliary small thermometer is clipped to the stem of the main thermometer. This auxiliary thermometer reads the temperature of the unimmersed thread. The correction, in degrees Fahrenheit, is then calculated from the following formula:

$$\text{Correction} = + 0{\cdot}0001\,(T - t)n,$$

where T, in degrees Fahrenheit, is the temperature read on the main thermometer; t, in degrees Fahrenheit, the temperature read on the auxiliary thermometer; and n the number of degrees Fahrenheit exposed on the main thermometer.

Depression of the Zero.—A far more serious error, in that it is less calculable, is what is called "the depression of the zero". If glass is subjected to a cyclic change of temperature it rarely returns to exactly the volume from which it started. Suppose that a thermometer is used

to take a reading at 212° F., and is then used to take a reading of the temperature of melting ice. The reading will be too low because of this hysteretic property of the glass of the thermometer.

For a similar reason, if a new thermometer is used to measure the temperature of, say, superheated steam, the temperature read will gradually fall if the true temperature of the steam is constant. This is because the glass does not fully expand at once. It may take several weeks before the glass has completely expanded and the temperature read will go on falling slightly so long as this phenomenon lasts. In engineering work this defect can often be got over fairly easily. For instance, in taking the temperature of the superheated steam supplied to a turbine, it is easy to have one thermometer constantly in each pocket. This thermometer becomes thoroughly "soaked", as it were, at the normal temperature of working, and after it has been in the pocket for, say, a month or so, a check reading can be taken against a standard instrument and the secondary thermometer can then be relied upon. There are several other corrections to which the mercury thermometer is subject, but these do not usually affect the reading sufficiently to be of much importance in engineering practice.

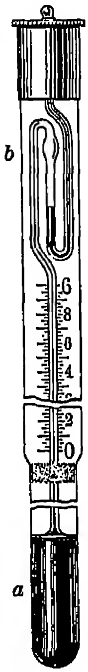

Fig. 5

The Beckmann Thermometer.—The engineer comes across the Beckmann thermometer in determining the calorific values of fuel. It is a special thermometer designed to read a small change of temperature very accurately. There are two bulbs—the usual bulb *a* and the upper bulb *b* (fig. 5). Suppose the instrument is to be used between 65° F. and 70° F., the bulb of the thermometer is carefully warmed up to 70° F., and mercury runs over into the top bulb. On cooling, the thermometer thread is broken at the top bulb, leaving a small quantity of free mercury in the top bulb. The remainder of the mercury is continuous with that in the bottom bulb. On cooling to 65° F. the end of the thread of mercury will be towards the bottom of the scale of the thermometer, if it is a 10° F. scale. Beckmann thermometers are usually calibrated in the centigrade scale, as they are essentially "scientific" instruments.

The bore of the thermometer is exceedingly fine, while the bulb *a* is comparatively large, so that the scale is a very open one. The movement of the end of the mercury thread is about 1 in. for a change of temperature of 1 degree C. The scale is divided into $\frac{1}{100}$° C. The thermometer is very sensitive, and its sensitiveness can be shown by a very simple and convincing experiment. If the bulb is taken suddenly but gently between the fingers and the thumb, the thread of mercury begins to shoot *down*, and immediately afterwards shoots up, assuming the instrument to be arranged for reading ordinary atmospheric temperatures. This is because the first thing that happens when the fingers warm the bulb is that the glass itself expands before the heat

gets to the mercury, and consequently the end of the thread drops, relative to the scale. Immediately the mercury begins to get heated, it recovers the lost ground and spurts up as it begins to take up the comparatively large supply of heat which it receives from the fingers.

CHAPTER III

Calorimetry

Calorimetry is the measurement of quantity of heat as distinguished from thermometry or pyrometry, which is concerned only with measurement of temperature.

The Unit of Heat has already been defined as the heat energy required to raise the temperature of 1 lb. of water 1 degree F.

Specific Heat.—It is found by experiment that the number of B.Th.U. required to raise 1 lb. of a given material 1° F. is a property of the material; for instance, it is about 0·09 for copper. Tables have been drawn up of this number for different substances (see Kaye and Laby, *Physical and Chemical Constants*). The number is called the *specific heat* of the substance. For solids and liquids the number is practically the same whether the substance is or is not allowed to expand in heating up. For gases and vapours, however, we have to distinguish between the *specific heat at constant pressure* and the *specific heat at constant volume.* The former (C_p) is as much as 40 per cent greater than the latter (C_v) for gases. The reason for this difference will be gathered from p. 190.

Suppose M is the mass of a substance A, which is heated from T_1° F. to T_2° F. Suppose S is its specific heat, T_1 the initial temperature, T_2 the final temperature, then the heat Q added to the body A is given by:

$$Q = MS(T_2 - T_1) \text{ B.Th.U.}$$

If we write Q_1 for the heat in the mass M at T_1, and Q_2 at T_2, we get:

$$Q_2 - Q_1 = MS(T_2 - T_1) \dots\dots\dots\dots (1)$$

It is usual to reckon the heat energy of a body from 32° F., i.e. we *suppose* that $Q_1 = 0$ when $T_1 = 32°$ F. This convention merely fixes a datum line, as it were. Equation (1) then becomes

$$Q_2 = MS(T_2 - 32) \dots\dots\dots\dots (2)$$

The principle employed in calorimetric measurements is contained in this equation.

Example.—Suppose we wish to find the temperature of a furnace. We can put a piece of metal (which does not melt in the furnace) of mass

M into the furnace until it takes up the temperature of the furnace. Let T be this temperature and S_1 the specific heat of the metal chosen, e.g. if it is platinum it will be about 0·04. This figure is got from a book of tables (Kaye and Laby's *Physical and Chemical Constants*, e.g.). The heat in the mass M, measured from 32° F., is then

$$MS_1(T - 32),$$

where S_1 is the specific heat of the metal. This body is quickly passed into a calorimeter, which is simply a vessel containing water and arranged to cool very slowly. Suppose the calorimeter is of copper and has mass m_1 lb., and has m_2 lb. of water in it. Suppose also that *before* the hot metal was placed in it, it was at t_o° F., then the heat in the calorimeter is

$$m_1S_2(t_o - 32) + m_2(t_o - 32),$$

where S_2 is the specific heat of copper. Suppose t is the final temperature of the calorimeter and contents *after* the hot metal ball has been put in the calorimeter, then the total heat of the apparatus is

$$MS_1(t - 32) + m_1S_2(t - 32) + m_2(t - 32).$$

The total heat of the system, consisting of the hot platinum and the calorimeter before and after mixing, must be the same if a negligible amount of heat has been lost by the apparatus in the process, hence

$$\begin{aligned} MS_1(T - 32) + m_1S_2(t_o - 32) + m_2(t_o - 32) \\ = MS_1(t - 32) + m_1S_2(t - 32) + m_2(t - 32). \\ \therefore MS_1(T - t) + m_1S_2(t_o - t) + m_2(t_o - t) = 0. \end{aligned}$$

But $(t_o - t)$ is negative, as $t_o < t$.

$$\begin{aligned} \therefore MS_1(T - t) &= m_1S_2(t - t_o) + m_2(t - t_o) \\ &= (m_1S_2 + m_2)(t - t_o). \\ \therefore T - t &= \frac{(m_1S_2 + m_2)(t - t_o)}{MS_1}, \\ \text{i.e. } T &= \left[\frac{(m_1S_2 + m_2)(t - t_o)}{MS_1}\right] + t. \end{aligned}$$

By observing M, m_1, m_2, t, and t_o by experiment we can calculate T from this formula, since S_1 and S_2 are given in tables of *specific heats*.

Experiment.—An iron ball weighing 0·1555 lb. is put into a furnace the temperature of which is less than the melting-point of iron. A calorimeter is fitted up which weighs 1 lb., is of copper, and contains 1 lb. of water. The initial temperature of the apparatus is 60° F. The ball is then quickly removed from the furnace and placed in the calorimeter.

The final temperature is 100° F. What is the temperature of the furnace?

$$\text{Here } S_2 = 0{\cdot}09,$$
$$S_1 = 0{\cdot}14,$$
$$t_o = 60,$$
$$t = 100,$$
$$m_2 = 1,$$
$$m_1 = 1.$$
$$\therefore m_1 S_2 + m_2 = 1{\cdot}09.$$
$$(t - t_o) = 40.$$
$$\therefore T = \left[\frac{1{\cdot}09 \times 40}{0{\cdot}1555 \times 0{\cdot}14}\right] + 100$$
$$= 2000 + 100$$
$$= 2100° \text{ F}.$$

This example has been given to illustrate the principle of calorimetry. It is not a good method of pyrometry because of (1) the uncertainty of the specific heats of metals at high temperatures; (2) the loss of heat in bringing the hot metal out of the furnace.

Calorific Value of Fuels.—Of the many instances in which heat measurement is required, it is only necessary here to deal with the measurement of heat evolved by combustion, that is, with the determination of the calorific power of fuel.

* Various forms of apparatus have been devised, from the time of Count Rumford till to-day, and they depend on the combustion of a known weight of the fuel in oxygen, and the cooling of the products of combustion in a known weight of water. Some of the forms of apparatus have been designed only for extremely accurate work in the research laboratory, whilst others are intended for practical purposes.

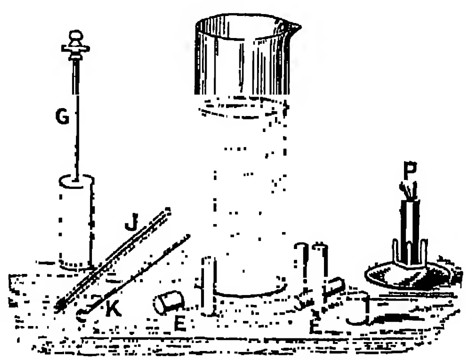

Fig. 6.—Thompson's Calorimeter

Thompson's Calorimeter.—This instrument is illustrated in fig. 6, and is intended for technical work, for which purpose it is admirably suited, and it has now come into general use. Its accuracy, however, is questionable. The fuel is not burned in oxygen, but in a mixture of potassium chlorate and nitrate which readily gives up its oxygen.

It consists of a glass vessel graduated to contain 2000 gm. of water, a copper cylinder E capable of holding the mixture of 2 gm. of fuel, with

* The rest of this chapter is taken, with slight modifications, from *Fuel and Refractory Materials*, by Sexton and Davidson (Blackie), by permission of the Publishers.

the necessary amount of fusion mixture; a copper base, on which the cylinder can be placed, and a copper cylinder G, with a row of holes round the bottom, to be placed over it. This cylinder is held in place by a set of springs on the base, and is also provided with a tube which reaches above the surface of the water, and which is fitted with a stop-cock.

Two grammes of the fuel will require from 20 to 24 gm. of the fusion mixture.* The mixture is put into the cylinder, tapped down, and placed on the base; a piece of slow match is put on the top and lighted; the cover G is quickly placed over it, and the whole is quickly put into the jar of water, combustion takes place, and the products of combustion pass up through the water and escape with a dense white smoke. Combustion once started should proceed vigorously and should be complete in about two minutes; the stop-cock is then opened so as to admit water into the interior of the cylinder. This is raised quickly up and down once or twice so as to thoroughly mix the water, and the temperature is read with the thermometer J. The temperature having been taken before the experiment, the rise of temperature in F. degrees × 1000 gives the number of B.Th.U. absorbed by the water.

There are various sources of loss—heat of decomposition of the fusion mixture, heat absorbed in warming the apparatus, and heat lost by radiation. To compensate for the last-named loss a blank experiment is first made, and then a second experiment, the water being cooled before starting about as much below the atmospheric temperature as it will be above it after. To allow for other sources of loss an addition of 10 per cent is made, which is said by the makers of the instrument to cover them.

The glass vessel is usually also graduated to contain 1932 gm. of water, and if that quantity be used, the reading gives at once the evaporative power of the fuel.

Temperature after experiment ..	61·5
" before experiment ..	47·5
	14·0
+ 10 per cent ..	1·4
	15·4 × 1000 = 15400

The quantity of combustion mixture required will vary with the nature of the fuel being treated, and charcoal, coke, or similar fuels should be burnt in a shorter and wider copper " furnace ".

W. Thomson's Oxygen Calorimeter.—In this apparatus the fuel is burned in oxygen. The apparatus (fig. 7) consists of a glass jar *a* capable of holding 2000 gm., or any other convenient known weight of water, a platinum crucible *g* in which the fuel is burned, and which rests on a clay cylinder; a bell-glass *f* which covers the crucible and contains the atmo-

* The best fusion mixture is 3 parts chlorate of potash and 1 part nitre; both it and the fuel should be quite dry. The slow match is readily made by soaking ordinary wick in a solution of lead nitrate and drying.

sphere of oxygen in which the combustion takes place. It is provided at the top with a neck, through which passes a glass tube *i*, by which the oxygen enters; it is provided with a stop-cock *k*, and can be raised or lowered as required. It rests on a perforated base, and is surrounded by a series of rings of wire-gauze to break up the ascending current of gas. A thermometer *d* and a stirrer are suspended in the outer vessel of water.

It is necessary to ascertain the heat capacity of the apparatus—that is, the amount of extra water to which the absorptive power of the apparatus is equal—which can be done once for all.

Two thousand grammes of water, at about 25° F. above the temperature of the air, is poured into the apparatus, and the water is well mixed. The water is left about the time which will be occupied by an experiment, and the amount by which the temperature is lowered gives the data for calculating the absorptive power of the apparatus sufficiently nearly for practical purposes.

Thus, to take an example:

Temperature of room, and therefore of the calorimeter before water is added	84° F.
Temperature of water	95° „
Temperature of apparatus and water after experiment	94° „

Therefore the fall of 2000 gm. of water 1° has raised the temperature of the calorimeter 10°, and the heat capacity of the apparatus is $\frac{2000}{10} = 200$ gm. of water.

This weight must therefore be added to the amount of water used and the 2000 gm. calculated as 2200 gm.

To make a determination.

About 1 to 1·5 gm. of the powdered fuel is carefully weighed and placed in the crucible, a short piece of slow match or an ignited vesta is placed on it, and the bell-glass is inverted over it. The whole is gently lowered into the water, and the oxygen is turned on, the delivery tube being pushed down to near the fuel if necessary. Combustion at once begins. When combustion is over, the oxygen apparatus is disconnected, and the stop-cock is opened so as to admit water, the bell-glass is moved up and down vigorously to ensure perfect mixture, and the temperature is taken.

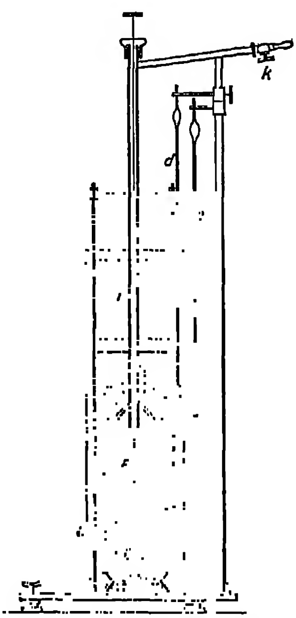

Fig. 7.—Thomson's Oxygen Calorimeter

Suppose 1 gm. of fuel were taken, and the rise of temperature of the water was to be 0·9°. Then since the 2000 gm. of water is equivalent

to 2200 gm., taking into account the absorptive power of the apparatus, the calorific value is

$$\frac{\cdot 9 \times 2200}{1} = 1980.$$

This apparatus gives fairly good results.

Several modified forms of this apparatus with improvements are now on the market, the best known being those of Rosenhain & Darling. In any case the oxygen used should be cooled to the atmospheric temperature before being passed into the combustion chamber.

The Berthelot-Mahler Calorimeter.—This is a form of bomb calorimeter which is the only really satisfactory form of calorimeter for

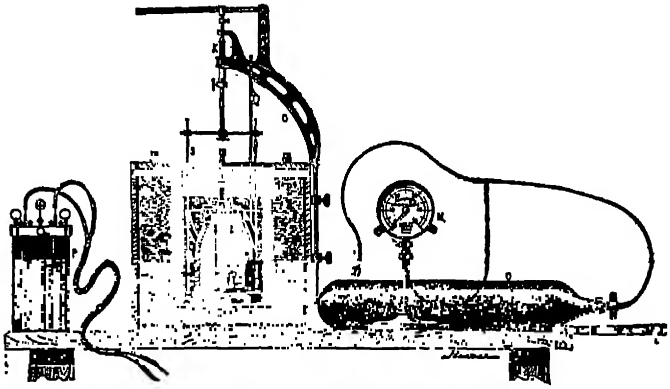

Fig. 8.—Berthelot-Mahler Calorimeter

solid fuels. It is a modification of the bomb used by Berthelot in his classic researches on the heat of combustion of different substances.

The combustion chamber B (fig. 8) is of mild steel. It is 8 mm. thick, and has a capacity of about 600 c. c. It is enamelled inside and nickel-plated outside, and is provided with a cover in which is a valve for the admission of oxygen. Wires also pass through the cover, by which a short coil of iron, or, better, platinum can be heated to ignite the fuel, which is contained in a platinum or silica crucible or capsule C. The whole is contained in a vessel of water D, provided with a spiral agitator S, and this again is placed in an outer vessel of water A.

The fuel is weighed in the crucible, which is then fixed in position; the igniting wire is weighed and adjusted, and the top is screwed down. Oxygen is allowed to enter from the cylinder O till the pressure on the gauge M indicates 25 atmospheres. The temperature of the water is noted at intervals of 1 min. for 5 min., an electric current is then passed from the battery P, and combustion takes place instantly. The temperature of the water is now taken at intervals of half a minute until the maximum is reached, then observations are continued at intervals of a minute for another 5 min., the stirrer being kept going all the time.

After the observations are completed the combustion vessel or "bomb"

is washed out, and any nitric acid present may be determined volumetrically if great accuracy is required.

It is necessary to determine the correction due to the loss of heat from the calorimeter during the test. The loss can be calculated by noting the rate at which the temperature rises or falls before firing and falls after firing. The addition to be made to the observed rise is generally not more than 2 per cent and can be calculated exactly from the periodical thermometer readings. The thermometer should be a very sensitive one, preferably a differential (Beckmann) thermometer, readable to 0·001° C.

Then if Δ is the rise of temperature, corrected,
W „ the weight of water in the calorimeter,
W′ „ the water equivalent of the apparatus, which must be determined by experiment,
n „ the weight of nitric acid, HNO_3,
f „ the weight of the spiral of iron wire,
0·23 „ heat of formation of 1 gm. of dilute nitric acid,
1·6 „ heat of combustion of 1 gm. of iron,
x „ the calorific power required,

$$\text{then } x = \Delta (W + W') - (0{\cdot}23n + 1{\cdot}6f).$$

Fig. 9 shows the Mahler-Cooke bomb which embodies certain mechanical improvements.

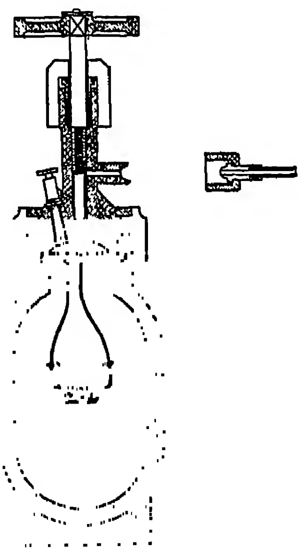

Fig. 9.—Mahler-Cooke Bomb

Calorimeters for Fuel Gas.—The Thomson oxygen calorimeter may be used for fuel gas; so can the Berthelot-Mahler form. In the latter the bomb is filled with gas before the oxygen is introduced, the pressure of oxygen required for coal-gas being about 5 atmospheres, and for producer-gas about 1·5 atmospheres.

Junker's Calorimeter.—This is one of the latest and best calorimeters for finding the calorific value of gases. A definite volume of gas is burnt, the amount consumed being measured by means of a meter, and a stream of water is kept flowing steadily through the apparatus. The volumes of gas and water, and the rise of temperature of the latter, supply the necessary data for calculating the calorific power.

A flame, 28 (fig. 10), is introduced into a combustion chamber formed by an annular copper vessel, the annular space being traversed by a number of copper tubes, 30. The heated gases circulate inside the tubes from the top to the bottom, whilst the

current of water travels outside the tubes in the opposite direction, all the heat produced by the flame being thus transferred to the water, the spent gases escaping at the atmospheric temperature. The pressure of the water is kept constant by two overflows, 3 and 20, and the quantity of water is regulated by the stop-cock, 9. A baffle-plate, 14, at the lower end of the apparatus, secures an even distribution of the water. The water can be passed through the tube 21, into a measured receptacle. To prevent loss by radiation the apparatus is enclosed in a nickel-plated cylinder. In addition to the calorimeter a meter capable of passing $\frac{1}{10}$ c. ft. for one revolution of the pointer, a water-supply giving 1 to 3 litres per minute, and two measure-glasses containing respectively 2 litres and 100 c. c. are required. The quantity of gas burned should be regulated so as to give out about 1000 to 1500 calories per hour (4000 to 6000 B.Th.U.); this is for illuminating gas 4 to 8 c. ft., or producer-gas 16 to 32 c. ft.

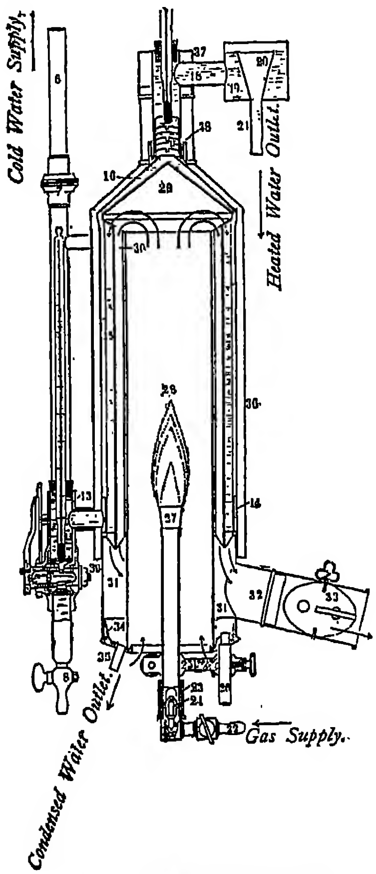

Fig. 10.—Junker's Gas Calorimeter

The gas is lighted, the thermometer placed in position, and the water turned on. The temperature rises, and the mercury soon becomes stationary. As soon as the temperature is steady, the hot-water tube is shifted over the large measure-glass. As the water flows, the temperature indicated by the thermometer is noted from time to time. As soon as 2 litres of water have passed, the gas is turned off, and the quantity of gas which has passed is read. The following is an example given by the makers of the instrument:

Meter Reading.	Cold-water Thermometer.	Hot-water Thermometer.	Water.
5 c. ft.	8·77° C.	26·75° C.	
	..	26·76	
	..	26·82	
	..	26·80	
	..	26·75	
5·344 c. ft. ..	8·77	26·80	2 litres
Mean ..	8·77	26·77	2 litres

To be strictly accurate, the water should be weighed, as 1 c. c. is not exactly equal to 1 gm.

If H is the calorific value of 1 c. ft of gas,
W „ quantity of water heated, in litres,
T „ difference in temperature between the hot and cold water,
C „ cubic feet of gas burned, then

$$H = \frac{WT}{C};$$

or in the case given,

$$H = \frac{2 \times 18}{\cdot 344} = 104{\cdot}65.$$

When it is desired to know the net calorific value, the quantity of water condensed should also be measured, as its latent heat must be deducted where the temperature of the products of combustion will, as in most cases, remain at a temperature above 100° C. The condensed water is drawn off by 35 into a measure-glass. In this case there was 53 c. c. of water. The latent heat of each cubic centimetre may be taken as ·6, so that the latent heat of the condensed water would be

$$\frac{\cdot 6 \times 53}{2} = 15{\cdot}9,$$

which must be deducted from the value obtained above, leaving 88·75 calories as the calorific power. Fig. 11 shows the new type of Junker's calorimeter.

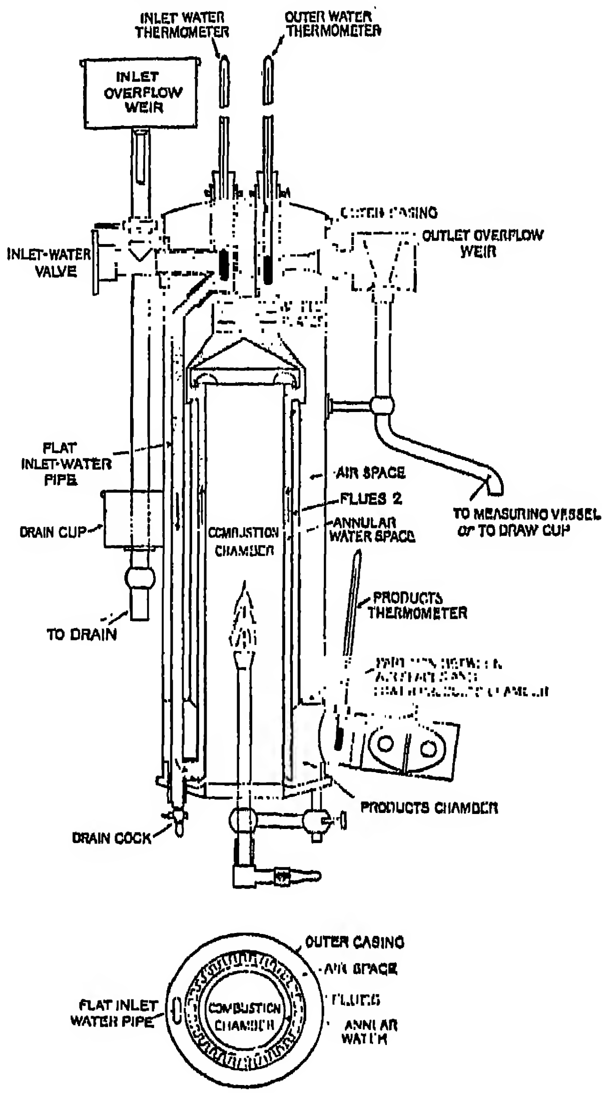

Fig. 11.—Junker's Gas Calorimeter (new type)

Boys' Gas Calorimeter.—This is another very good type of gas

calorimeter which gives results differing from those of the Junker's calorimeter by not more than a quarter of 1 per cent. It is the instrument prescribed for gas-testing in London by the Metropolitan Gas Referees, whose description is as follows:

"The gas calorimeter which has been designed by Mr. Boys is shown in vertical section in fig. 12. It consists of three parts, which may be separated, or which, if in position, may be turned relatively to one another about their common axis. The parts are (1) the base A, carrying a pair of burners B, and a regulating tap. The upper surface of the base is covered with a bright metal plate held in place by three centring and lifting blocks. The blocks are so placed as to carry (2) the vessel D, which must rest upon the horizontal portion of the blocks and not upon their upturned ends. The vessel is provided with a central copper chimney E and a condensed-water outlet F. It is jacketed with felt R, protected by a sheet of metal S. The diameter of the chimney E at the base is $3\frac{1}{4}$ in. and at the top $2\frac{1}{2}$ in. and its thickness $\frac{1}{50}$ in. The base of the outer vessel

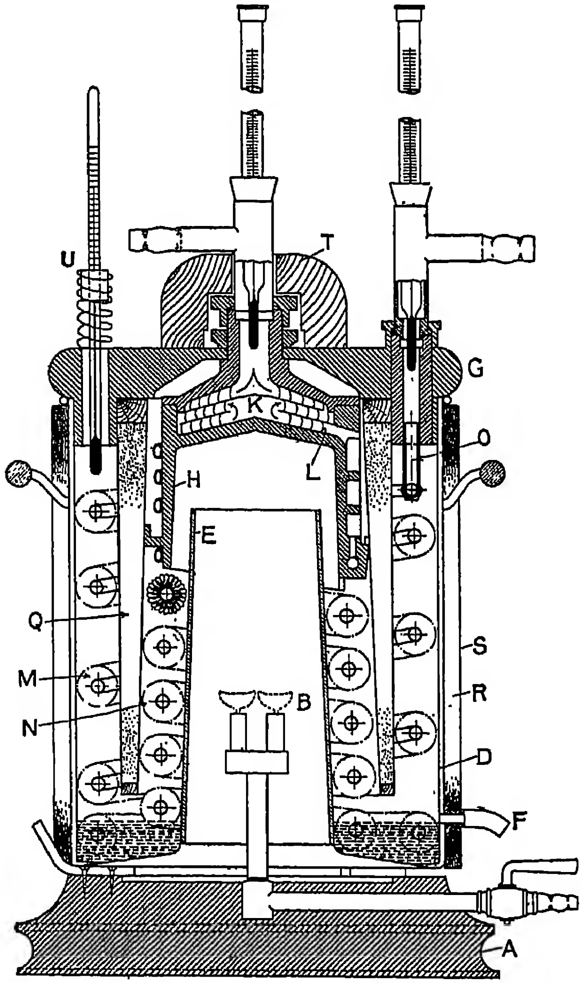

Fig. 12.—Boys' Gas Calorimeter

shown in the drawing as a separate piece is preferably spun in one piece with the chimney. In order to prevent obstruction of the flow of condensed water from the outlet F by the accidental contact of the thin brass protecting wall of the pipe system, a small dimple is punched in the outer casing on either side of the outlet F so as to project inwardly about $\frac{1}{16}$ in. Resting upon the rim of the vessel D are (3) the water-circulating system of the calorimeter attached to the lid G. Beginning at the centre, where

the outflow is situated, there is a brass box which acts as a temperature equalizing chamber for the outlet water. Two dished plates of thin brass K, K are held in place by three scrolls of thin brass L, L, L. These are simply strips bent round like unwound clock springs, so as to guide the water in a spiral direction inwards, then outwards, and then inwards again to the outlet. The lower or pendant portion of this box is kept cool by circulating water, the channel for which may be made in the solid metal, as shown on the right side, or by sweating on a tube, as shown on the left. Connected to the water channel at the lowest point by a union are five or six turns of copper pipe such as is used in a motor-car radiator of the kind known as Clarkson's. In this a helix of copper wire threaded with copper wire is wound round the tube, and the whole is sweated together by immersion in a bath of melted solder. A second coil of pipe, of similar construction, surrounding the first is fastened to it at the lower end by a union. This terminates at the upper end in a block, to which the inlet water box and thermometer holder are secured by a union as shown at O. An outlet water box D and thermometer holder are similarly secured above the equalizing chamber H. The lowest turns of the two coils M, N are immersed in the water which in the first instance is put into the vessel D.

"Between the outer and inner coils M, N is placed a brattice Q made of thin sheet brass, containing cork dust to act as a heat insulator. The upper annular space in the brattice is closed by a wooden ring, and that end is immersed in melted rosin and beeswax cement to protect it from any moisture which might condense upon it. The brattice is carried by an internal flange which rests upon the lower edge of the casting H. A cylindrical wall of thin sheet brass, a very little smaller than the vessel D, is secured to the lid, so that when the instrument is lifted out of the vessel and placed on the table, the coils are protected from injury. The narrow air space between this and the vessel D also serves to prevent interchange of heat between the calorimeter and the air of the room.

"The two thermometers for reading the water temperatures and a third (U) for reading the temperature of the effluent gases are all near together, and at the same level. The thermometer U, divided on the Fahrenheit scale, is supported as shown in fig. 12 by means of a cork and an open spiral of wire, so that the bulb is a short way above the circulating coil, and with its stem passing through one of the five holes provided for the effluent gases. The lid may be turned round into any position relatively to the gas inlet and condensed-water drip that may be convenient for observation, and the inlet and outlet water boxes may themselves be turned so that their branch tubes point in any direction. A wood shield T, made in two halves, serves to protect the outlet water box from loss of heat.

"A regular supply of water is maintained by connecting one of the two outer pipes of the overflow funnel to a small tap over the sink. The overflow funnel is fastened to the wall about 1 m. above the sink, and the other outer pipe is connected to a tube in which there is a

diaphragm with a hole about 2·3 mm. in diameter. This tube is connected to the inlet pipe of the calorimeter. A piece of stiff rubber pipe, long enough to carry the outflow water clear of the calorimeter, is slipped on to the outflow branch and the water is turned on, so that a little escapes by the middle pipe of the overflow funnel and is led by a third piece of tube into the sink. The amount of water that passes through the calorimeter in four minutes should be sufficient to fill the graduated vessel shown in fig. 13 to some point above the lowest division, but insufficient in five minutes to come above the highest division. If this is not found to be the case, a moderate lowering of the overflow funnel or reaming out of the hole in the diaphragm will make it so. The overflow funnel should be provided with a lid to keep out dust. The graduated vessel (fig. 13) shall have been previously examined and certified by the Gas Referees. An impression of the stamp, on the base of the vessel, shall be accepted as proof that the vessel has been thus certified.

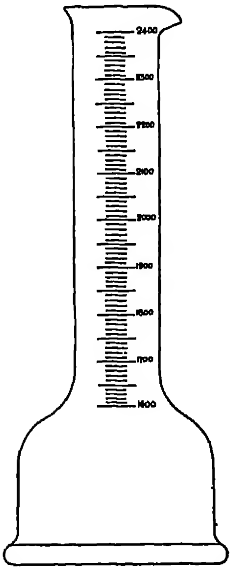

Fig. 13.—Graduated Vessel

"The thermometers for reading the temperatures of the inlet and outlet water should be divided on the centigrade scale into tenths of a degree, and they should be provided with reading lenses and pointers that will slide upon them. The thermometers are held in place by corks (not india-rubber), making an air-tight fit within the inlet and outlet water boxes. Care must be taken that the bulbs are fully immersed."

There are several forms of recording gas calorimeters on the market. Some of these are not at all reliable. Others are very expensive and short lived. The Gas Regulation Act, 1920, makes their use almost imperative.

Calorimeters for very Volatile Liquids.—For very volatile liquid fuels, such as petrol, the ordinary forms of calorimeter are, if not inapplicable, at least unsatisfactory. In such purposes a modified form of gas calorimeter is best suited, the fuel being burnt in a lamp.

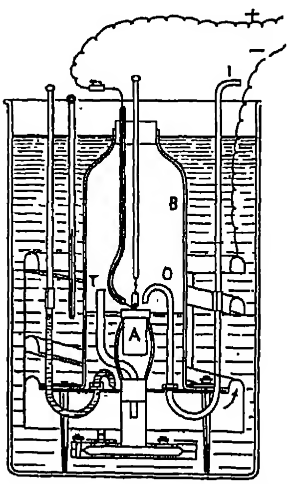

Fig. 14.—"Darling" Calorimeter

The "Darling" calorimeter is shown in fig. 14. The fuel is burnt in the lamp A, which is fitted with an asbestos wick, and which has a capacity of 3 or 4 c. c. The lamp is burnt in the bell-jar B, and is ignited electrically. The oxygen required is introduced by the tube I, and enters through the copper tube O, impinging on the top of the lamp. The

products of combustion pass away by the tube T and bubble up through the water through holes in the perforated base plate H. For very volatile liquids, such as petrol, the lamp is surrounded by cold water during combustion.

This form of apparatus, or an adaptation of the ordinary gas calorimeter, gives satisfactory results.

Comparison of Calculated and Determined Results.—For various reasons the results calculated from the various formulæ are not exactly correct, the variation being sometimes one way and sometimes the other. The formulæ are based on the assumption that the elements evolve the same amount of heat, when burnt in the condition of combination in which they exist in the fuel, that they would do if burnt in the free condition. This assumption is clearly a doubtful one. If the bodies in which they exist were formed with the evolution of heat, the results calculated from ultimate analysis will be too high; if with absorption of heat, the calculated results will be too low. In the case of solid fuels, the form of combination in which the elements are present is quite unknown, but probably they are comparatively unstable, and formed with but little heat change. The same is true in the case of liquid fuels, but, in the case of a gaseous fuel, the proximate composition of which is known, the calculated figures should be more accurate. A determination of the heating power by means of a calorimeter is always preferable to a calculation.

Of the various types of calorimeter in use for solid and liquid fuels the bomb is decidedly the most accurate, and it is generally used where accuracy is required. It is, however, rather expensive and perhaps too complicated for ordinary technical use. Next to it stand the oxygen calorimeters, which, when carefully used, give good results, whilst the most generally used instrument, the Thompson calorimeter, with a fusion mixture, is not to be depended on except for rough approximations.

CHAPTER IV

Combustion*

Combustion.—Heat is required in almost all engineering operations and is always obtained directly or indirectly by the combustion of substances called fuels.

Chemical combination is almost invariably attended with the evolution of heat, and when this is sufficient to raise the combining bodies or the products of combination to the temperature at which they evolve light, combustion is said to take place. Combustion may therefore be defined

* This chapter is taken, by permission, with slight modifications, from *Fuel and Refractory Materials*, by Sexton and D[illegible].

as vigorous chemical combination, attended with the evolution of light. In practice the combination is always between a combustible or fuel and the oxygen of the air, which is therefore said to support combustion.

Conditions which favour Combustion.—In order that combustion may begin, the fuel must be brought in contact with the air, at a suitable temperature; and in order that it may continue, this temperature must be kept up, a supply of oxygen must be maintained, and the products of combustion must be removed.

As a rule, the larger the surface of the combustible in contact with the air, the more readily will combustion take place; and as gases, by the mobility of their molecules, allow of the largest possible amount of contact, they, if combustible, usually burn very readily.

If a combustible gas be allowed to escape from a tube into the air at a temperature at which it will burn, combustion takes place with great facility; for as the gas comes into the air diffusion takes place, the gas molecules are brought into close contact with the oxygen molecules, and they combine, forming a zone of combustion surrounding a core of gas, and thus producing a flame.

If a combustible gas be thoroughly mixed with air and a light be applied, combustion takes place almost instantly through the whole mass, travelling very rapidly from particle to particle; heat is suddenly evolved, great expansion results unless it is mechanically prevented, and an explosion of more or less violence takes place.

Liquids do not burn readily in mass, for the air cannot penetrate them, and there is therefore only contact at the comparatively small surface of the liquid. There are apparent exceptions to this, due to the fact that most liquids are volatile, and combination therefore takes place near the liquid surface between the vapour and the air. If a combustible liquid be broken up into a fine spray by a steam or air jet, it will burn almost exactly as if it were a gas, and will form an explosive mixture with air.

Combustible solids usually burn readily when in pieces of such a size as to allow ready access of air, and at the same time expose a large surface of contact. If the lumps be too large the contact surface is too small and combustion is hindered, and if the substance be in a powder the air will be unable to penetrate, and therefore there will still only be a small surface of contact. If a finely-powdered solid fuel be projected at a high temperature into air, it burns very rapidly, almost exactly in the same way as a gas, and such a powder may even form an explosive mixture with air. It is quite certain that many colliery and other explosions, if not entirely due to, are at any rate very much intensified by the presence of coal or other combustible dust in the air.

Many examples of the influence of contact surfaces are familiar. A piece of charcoal of large size will burn readily, because, as it is very porous, air can find its way into it, and thus provide a large contact surface, while a large lump of anthracite, not being porous, will hardly burn at all. Paper and wood-shavings are employed to light a fire, though they have almost

the same composition as the wood, because, owing to their thinness, they expose a large surface to the air, and ignite readily. These and such materials as light fabrics are very combustible, but books in which the paper leaves are pressed closely together, and bales of fabrics, are very difficult to burn; and when a warehouse has been burned, whilst all the loose goods are destroyed, it is quite common to find bales of the same materials only singed on the outside. Heavy beams of wood make fireproof floors, and sawdust or coal-dust thrown on a fire will often extinguish it.

Proportion of Combustible.—In order that combustion may take place, the combustible and air must be present in—within certain limits—definite proportions. This is not so noticeable in the case of solid or liquid fuels, or of gas burned in a flame, because, owing to the circulation set up in the air, the proportions to some extent adjust themselves. It is, however, well seen in the case of mixtures of gases. If coal-gas and air be mixed in certain proportions a violently explosive mixture results; but there may be a large quantity of gas in the air, enough to be detected by the smell and to produce chemical and physiological effects, and yet the mixture will not explode on applying a light. The presence of a large quantity of inert matter in a fuel may much hinder or even prevent combustion, whilst the presence of a comparatively small quantity of carbon dioxide in the air will prevent it supporting combustion. Professor Clowes has recently shown that air which contains about 4 per cent of carbon dioxide, the oxygen being reduced by a like amount, will extinguish ordinary combustibles, such as candles or oil flames. Marsh-gas must be mixed with at least 17 times its own volume of air to form an explosive mixture.

Temperature of Combustion.—For combustion to take place a certain temperature, varying with the nature of the combustible, is necessary. A mixture of hydrogen and oxygen in explosive proportions will remain inert for any length of time until a portion of the mixture is raised to about 1100° F. (590° C.), when ignition will at once take place. Coal-gas ignites in air at a red heat. Many of the metals are not acted on by dry oxygen at ordinary temperatures, but if they be heated to redness some of them burn, as in the case of magnesium, with great brilliance. On the other hand, phosphorus ignites at such a low temperature that for safety it is always kept under water or otherwise protected from contact with the air, and some substances have such affinity for oxygen that they take fire on coming in contact with it.

The temperature at which combustion can take place varies also with the condition of the combustible. Lead and iron can be obtained in such a fine state of division that they take fire spontaneously in air at ordinary temperatures.

Continuous Combustion.—In some cases when a substance has been ignited it will continue to burn, in others it will go out as soon as the external source of heat is removed. This depends on the relationship which

exists between the heat evolved by combustion and the temperature of ignition. If the heat evolved be sufficient to maintain the temperature above the ignition point the combustion will continue, if not it will cease.

Combustibles and Supporters of Combustion.—The fuel or substance which burns is usually called a combustible, and the oxygen of the air is called a supporter of combustion. These terms, though convenient, are not strictly correct, except in so far as they indicate an accident of position. Combustion is a mutual action in which both substances play an equal part, and which of them is the combustible and which the supporter of combustion depends on circumstances. When a solid combines with a gas, the gas surrounds it, and is therefore regarded as a supporter of combustion. When a mixture of a combustible gas such as hydrogen and air is exploded it is impossible to say that either is the combustible rather than the other; but when gas is burnt at a jet the flame is surrounded by the excess of air, which is therefore called a supporter of combustion. It is quite simple to arrange experiments so as to burn air in coal-gas or oxygen in hydrogen, and thus reverse their usual positions.

Complete and Incomplete Combustion.—All combustibles in common use are composed chiefly of carbon (C), usually combined with hydrogen (H), oxygen (O), and sometimes small quantities of other elements, but the carbon and hydrogen are always the valuable constituents.

When a combustible burns, the combustion may be either complete or incomplete. It is complete when all the combustible constituents are oxidized to their highest state of oxidation, and it is incomplete when any fuel is either left unconsumed or passes away combined with less oxygen than the maximum with which it is capable of combining.

In the case of hydrogen, there is only one compound that can be formed —water, H_2O; and therefore, if the combustion be incomplete, some of the hydrogen must remain unconsumed.

In the case of carbon, the highest state of oxidation is carbon dioxide, CO_2; but there is also another oxide, carbon monoxide, CO, which contains, for the same amount of carbon, only one-half as much oxygen. When carbon is incompletely burned, therefore, either carbon may be left unconsumed, or carbon monoxide may be formed and pass away with the products of combustion. Both carbon dioxide and carbon monoxide are colourless gases, so that it is often not easy to decide whether combustion is complete or not.

The combustion of carbon begins at a comparatively low temperature; at about 750° F. the product is almost entirely carbon dioxide; as the temperature rises the rate of combustion increases, and the proportion of carbon monoxide formed increases till, at 1830° F., the product is almost entirely this gas, which, if the air supply be sufficient, is rapidly burned to carbon dioxide. It is for this reason that carbon at low temperatures simply smoulders, whilst at very high temperatures it burns with a flame.

The combustion of hydrocarbons is much more complex. If the

combustion be quite complete, the products are water and carbon dioxide; but if it be incomplete, the products which will be formed depend on circumstances. Many hydrocarbons dissociate or split up into simpler hydrocarbons with separation of hydrogen, as, for instance, ethylene, C_2H_4, which goes partly to hydrogen and acetylene, C_2H_2. The hydrogen burns to water, and the acetylene, partially escaping as such, imparts a most unpleasant odour to the products of combustion of incompletely-burnt hydrocarbon gases. The carbon will be burned to carbon dioxide, or a mixture of this and carbon monoxide. Under some conditions carbon may also be separated by dissociation in the solid form as soot.

Incomplete combustion of any kind always means considerable loss of heat.

Conditions of Complete Combustion.—In order to ensure complete combustion, three things are essential: the air supply must be sufficient, the air must be brought into intimate contact with the fuel, and the temperature must be kept up to ignition point until combustion is quite complete. Either insufficient air supply or too rapid cooling are the usual causes of incomplete combustion.

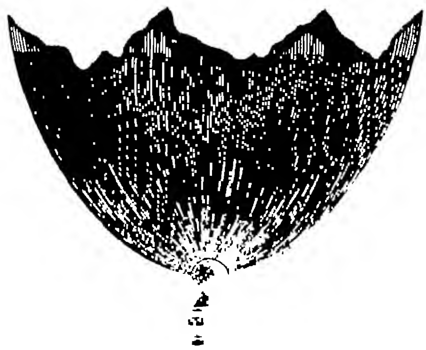

Fig. 15.—Gas Flame

Flame.—Fuels burn in very different ways. Some, as, for instance, charcoal at low temperatures, burn with a glow, evolving but little light; others, such as the metal magnesium, burn with a very brilliant light, but with no flame; others, like hydrogen, burn with a non-luminous flame; and lastly, some, like coal-gas, burn with a bright luminous flame. All combustible gases—and gases only—burn with a flame. There are some apparent exceptions to this, but they are only apparent; and whenever a solid or liquid seems to burn with a flame, it is because it is converted into gas either before or during combustion. "Flame is gas or vapour, the surface of contact of which with the atmospheric air is burning with the emission of light" (Percy). As combustion only takes place at the surface of contact, the flame must be hollow.

A simple flame is one in which there is only one product of combustion, and a compound flame (fig. 15) is one in which there are two or more. Almost all flames used in the arts are compound, the only examples of simple flames being those of hydrogen and carbon monoxide.

Simple Flame.—As an example of a simple flame a jet of hydrogen burning in air may be taken. As the hydrogen escapes from the jet it displaces the air and then diffuses into or mixes with it, and in the space where the gases mix combustion takes place. In the centre of the flame,

therefore, will be a core of unburnt gas, outside will be the air, and between is the zone of combustion, or the flame.

Compound Flames.—With compound flames the reactions are much more complex, and it is often difficult to determine the exact changes which take place. The compound flames of most practical importance are those obtained by the combustion of various hydrocarbon gases. The fuel may be used in the form of gas from a burner, in the solid form as a candle, or in the liquid form as in the case of the burning oils. In the last two examples the combustible is drawn up through the wick by capillary attraction; gasified by the heat evolved during combustion, and the gas is burned; indeed, candles and lamps may be regarded as combined gas producers and burners.

The structure of a compound flame is much the same as that of a simple one. In the centre is a core of unburnt gas, outside is the air, and between the two is the zone of combustion or the flame; but this zone is much more complex than in a simple flame, the combustion in the inner portions being usually incomplete.

Luminosity of Flame.—Most of the flames produced by hydrocarbons are more or less luminous, and the cause of this luminosity has given rise to a vast amount of discussion. Clearly the luminosity is not due to temperature alone, for a hydrogen flame, especially when formed by a mixture of hydrogen and oxygen, is intensely hot, but almost non-luminous; and it is quite possible to burn coal-gas, e.g. in the Bunsen burner, in such a way as to give a very hot but non-luminous flame.

Davy suggested that the luminosity of flame was due to the separation of particles of solid carbon by incomplete combustion within the flame, which, being heated to intense whiteness, evolve light. This theory has been generally accepted, and in its favour many facts may be quoted:

1. If solid matter be introduced into a hot, non-luminous flame light is evolved. A cylinder of lime, for instance, placed in an oxy-hydrogen flame gives the brilliant lime-light; a mantle of thoria and ceria suspended in a non-luminous Bunsen flame is the Welsbach incandescent burner.

2. If a cold surface be held in an ordinary luminous flame it becomes covered with a black deposit of solid carbon or soot.

3. Many substances which burn with the formation of solid products of combustion, e.g. magnesium or zinc, give an intense white light.

4. Luminous flames when examined with a spectroscope give a continuous spectrum.

5. When sunlight is reflected from hydrocarbon flames, it is polarized exactly in the same way as light reflected from solid carbon particles suspended in air.

On the other hand, Frankland contended that—at least in many cases—luminosity was due to the presence not of solid particles but of very dense gases or vapours. In support of this view it may be urged:

1. That the luminosity of many flames is much increased under pres-

sure, even a flame of hydrogen becoming luminous at high pressures, and that under reduced pressures the luminosity of ordinary luminous flames is much reduced.

2. That many substances, e.g. phosphorus and arsenic, burn with a very luminous flame, though at the temperature of combustion the products are gaseous. Hydrogen produces water having a vapour density of 9 (H = 1), and the flame is non-luminous. Phosphorus produces phosphoric anhydride, P_2O_5, having a density of 71, and the flame is luminous; and arsenic, which produces arsenious oxide, As_4O_6—vapour density 198—also burns with a luminous flame. Therefore the luminosity of the flame seems to depend on the density of the products of combustion.

3. Soot is not pure carbon, but always contains hydrogen; and further, the fact that soot is deposited does not prove that it existed as such in the flame, as it may have been produced by the decomposition of dense hydro-carbons.

4. Gases under great pressure give much more complex spectra than under ordinary conditions, becoming banded, and ultimately tending to become continuous.

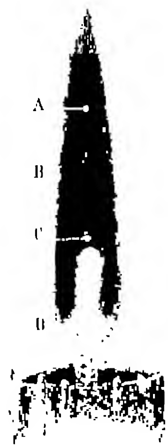

Fig. 16.—Candle Flame

For these reasons it has been urged that luminosity of ordinary hydrocarbon flames may be due to the presence of very dense hydrocarbons, which under suitable conditions split up into carbon and hydrogen; or light hydrocarbons, which are then burned.

Professors Lewes' and Smithells' Researches.—Professors V. B. Lewes and A. Smithells investigated the question of the luminosity of coal-gas and similar flames, and whilst their work confirms the view that the luminosity is due to separated carbon particles, it has thrown fresh light on the reactions within the flame by which these are separated and the conditions under which luminosity can be produced.

Prof. Smithells describes the structure of an ordinary luminous gas flame as consisting of four parts, which, for convenience, may be taken in the inverse order to that in which they are given by him:

(1) A dark inner core or region, consisting principally of unburned gas, mingled with some products of combustion which have diffused in from the surrounding parts; (C in fig. 16).

(2) A yellow luminous portion, marking the region in which hydrocarbons are undergoing decomposition, the heat producing the dissociation being largely derived from the outer zones; (A).

(3) An inner light-blue portion (D), visible at the base of the flame; and

(4) An outer sheath or mantle (B); these parts (1 and 2) corresponding

to the outer and inner flame-cones of the Bunsen burner, and marking the region where the gas is undergoing combustion in presence of excess of air.

The explanation which has usually been given of the phenomena of luminosity is something like this: The gas coming into the air is at first mixed with only a very limited supply of air; the hydrocarbons cannot be completely burned, therefore the hydrogen burns, forming water, and the carbon is liberated and heated to incandescence by the heat evolved by the combustion of the hydrogen.

This description has been shown to be incorrect in at least two points: (*a*) the order of combustion, and (*b*) the source of the heat.

(*a*) The hydrogen does not burn first, in the case of methane at any rate; the reaction being, as pointed out by Dalton, $CH_4 + 2O = CO + H_2O + 2H$—water, carbon monoxide, and hydrogen being thus formed. But as carbon monoxide and water can mutually decompose each other, $CO + H_2O = CO_2 + 2H$, a further reaction may take place till the system attains equilibrium, the conditions of which, according to Dixon, are expressed by the relation $\frac{C_{CO} \times C_{H_2O}}{C_{CO_2} \times C_{H_2}} = 4$. This is subject to certain conditions of temperature and dilution. These reactions, however, probably take place only to a small extent in the inner luminous part of the flame.

(*b*) The source of the heat, therefore, is to be sought not altogether in this partial combustion, but in transmission from the outer zone, where the temperature is very high.

The constituent of coal-gas to which the luminosity is mainly due is ethylene, C_2H_4, and perhaps some of its higher homologues. This at high temperatures (1500 to 1800° F.) splits up, yielding acetylene and methane, $3C_2H_4 = 2C_2H_2 + 2CH_4$, the acetylene then polymerizing into more complex hydrocarbons. At higher temperatures (above 2130° F.) no polymers are formed, but only acetylene, and at that temperature methane also dissociates, yielding acetylene and hydrogen, $2CH_4 = C_2H_2 + 3H_2$, so that all the hydrocarbons present will have split up into acetylene and hydrogen. At still higher temperatures acetylene itself splits up into carbon and hydrogen, this change taking place at about 2430° F.

The various hydrocarbons, as is well known, burn with very different degrees of luminosity. The flame of methane, CH_4, is very slightly luminous, that of ethylene, C_2H_4, is more luminous, whilst that of acetylene is intensely brilliant. That the luminosity is not due merely to the amount of carbon which the combustible contains is shown by the fact that in equal volumes of gas ethylene and acetylene contain the same weight of carbon, whilst benzene, C_6H_6, which burns with a luminosity much inferior to that of acetylene, contains, in the gaseous condition, three times as much carbon.

Neither is the luminosity due to temperature alone, as has been already shown.

It is now possible to form some idea of what actually takes place in a luminous gas flame. The inner blue core must be regarded as altogether unburned gas, for in it no combustion is taking place, but the temperature is rising, much heat being received by radiation from the outer zones of the flame, and the hydrocarbons being to some extent dissociated. As the temperature rises, dissociation takes place to a greater extent, till all the hydrocarbons, or nearly so, are converted into acetylene, and then the acetylene itself undergoes dissociation. This dissociation evolves heat, and at once brings the separated carbon up to vivid incandescence. As the products of these reactions pass outwards they are burned, the temperature therefore rising to the edge of the flame, and the combustion is completed in the very hot but feebly luminous mantle.

The luminosity, therefore, seems to depend not so much on the actual amount of carbon contained in the gas, or on the temperature, as on the readiness with which the gases present form acetylene or some other hydrocarbon which will similarly dissociate.

The luminosity of an acetylene flame is much reduced by the admixture of other gases, even though they themselves are combustible and evolve a large quantity of heat; as, for instance, hydrogen and carbon monoxide. The presence of such gases not only reduces the luminosity of the flame, but enormously raises its dissociation temperature, which is the point at which luminosity begins; thus:

Percentage of Acetylene.	Percentage of Hydrogen.	Temperature of Luminosity.
100	0	1440° F.
90	10	1650
80	20	1830
10	90	3090

The above "acetylene theory" of the chemical reactions on which luminosity of flame depends is not now generally accepted.* In any case the reactions are complex.

Non-luminous Combustion.—If coal-gas, or any other gas which usually gives a luminous flame, be burnt in such a way that excess of oxygen penetrates into every part of the flame, and the acetylene burns before it can undergo dissociation, the flame will be non-luminous. Thus, if a gas flame be turned very low it is non-luminous, so also is candle flame when the wick is very short. The best means of obtaining a non-luminous flame is the Bunsen burner.

The Bunsen Burner.—This burner consists of a tube, usually about $\frac{3}{8}$ in. in diameter and $3\frac{1}{2}$ in. long, though the size may be varied within wide limits. Gas is admitted to the bottom of the tube, and just above the jet by which the gas enters are holes for the admission of air. The air (usually $2\frac{1}{2}$ volumes) mixes with the gas, and the mixture burns with a hot non-luminous flame.

* W. A. Bone and H. F. Coward, *J. Chem. Soc.*, 1908, p. 1197.

"When a Bunsen burns under normal conditions it has a bluish central zone, but if the air supply be largely in excess of that required for non-luminous combustion, the flame becomes smaller and fiercer with the formation of a green central zone." *

The cause of the non-luminosity of the flame has usually been attributed to the more perfect combustion of the hydrocarbons due to the excess of oxygen in the interior of the flame. Professor Lewes has shown that this is not by any means necessarily the case, as nitrogen, carbon dioxide, and other inert gases also prevent luminosity; dilution, as already remarked, very much retarding the dissociation of acetylene, and therefore the production of luminosity. It is this dilution which is generally efficient in preventing luminosity in the Bunsen; but if the air supply be too large then oxidation takes place rapidly, and the inner cone changes in appearance. The temperature of the flame is a little higher when the diluent is air than when it is nitrogen, as the following table shows:

	Air.		Nitrogen.	
½ in. above burner	54° C.	129° F.	30° C.	86° F.
1½ " " "	175	347	111	231
Tip of inner cone	1090	1990	444	831
Centre of outer cone	1535	2795	999	1830
Tip of outer cone	1175	2150	1150	2100
Side of outer cone level with tip of inner cone	1335	2435	1235	2255

If the supply of air be too small then a luminous point appears at the tip of the inner cone.

It does not follow that because the flame is non-luminous, combustion is complete; it frequently happens that the ingress of air has cooled the gases below ignition point, and not "inconsiderable portions of methane, carbon monoxide, acetylene, and even hydrogen escape unburnt", both from non-luminous and luminous flames.

Professor Lewes thus describes the three zones of an ordinary luminous flame:

"1. The inner zone, in which the temperature rises from a comparatively low point at the mouth of the burner to about 1000° C. (1830° F.) at the apex of the zone. In this portion of the flame various decompositions and interactions occur, which culminate in the conversion of the heavier hydrocarbons into acetylene, carbon monoxide being also produced.

"2. The luminous zone, in which the temperature ranges from 1830° F. to 2370° F. Here the acetylene formed in the inner zone becomes decomposed by heat with liberation of carbon, which at the moment of separation is heated to incandescence by the combustion of the carbon monoxide and hydrogen, thus giving luminosity to the flame.

"3. The extreme outer zone. In this part of the flame, the combus-

* V. B. Lewes, *J. S. C. I.*, 1892, p. 231.

tible gas meeting air, combustion takes place, making this the hottest part of the flame; but towards the outer part of this zone, combustion being practically completed, the cooling and diluting influence of the entering air renders a thin layer of the flame non-luminous, finally extinguishing it. This description of a luminous flame is of necessity far from complete."

The various causes of loss of luminosity in a Bunsen burner are summarized by the same writer and include the following:

1. The atmospheric oxygen burns up the hydrocarbons before they can afford acetylene.

2. The presence of nitrogen increases the temperature necessary to bring about the partial decomposition of the hydrocarbons.

3. The cooling action of the air introduced.

The Lewes acetylene theory and that of preferential combustion have been discredited to a certain extent by the recent work of Bone and his collaborators. As a result of their researches, which had chiefly to do with the oxidation of methane, the "association" theory has been put forward as a more satisfactory explanation of the general phenomena of combustion. According to this theory, the hydrocarbons at first react with the oxygen of the air to form unstable hydroxylated products, which in the presence of sufficient oxygen decompose into water and carbon dioxide. Thus methane probably forms the following successive oxidation products: methyl alcohol, formaldehyde and steam, formic acid, carbonic acid, and finally carbon dioxide and water.

At best the reactions in combustion are as yet but little understood.

Propagation of Flame.—If a long glass tube, closed at one end, be taken and filled with an explosive mixture, say of coal-gas and air, and a light be applied at the open end, the flame will run down the tube with a definite and usually measurable speed, combustion not taking place instantaneously, but the ignition being transmitted from molecule to molecule at a comparatively slow rate. The speed at which the flame travels is called "the speed of propagation of the flame", and combustion thus taking place has been called an "explosion of the first order".

If, instead of a closed tube filled with gas, a tube open at both ends be used, and the mixture be made to flow through it, and if the gas be lighted so that the flame has to travel in the opposite direction to that in which the gas is flowing, its speed of transmission will be reduced, and will be the difference between the speed of propagation and the rate at which the gas is flowing, if the former be greater than the latter. If the rate of flow of the gas be very slightly in excess of the speed of propagation of the explosion, the flame will remain just at the mouth of the tube, and if it be much greater there will be a more or less long flame. Deville made a series of most interesting experiments on the rate of propagation of flames, and it is to his work, and that of Bunsen, that we owe most of our knowledge of the subject. He burnt a mixture of two volumes of carbon mon-

oxide and one volume of oxygen—the gases therefore being almost exactly in the proportions required for complete combustion—at a jet having an area of 5 sq. mm. A flame 70 to 100 mm. high was formed, which consisted of two portions, an inner cold core, 10 mm. high, and an outer flame zone. It is obvious that in this case the inner core was not due to the absence of oxygen for combustion, but to the fact that the gases were travelling forward at such a speed that the flame could not travel backwards and ignite the mixture in the tube; and no doubt had the rate of flow been diminished the flame would have grown smaller and ultimately lighted back.

When a light is applied to an explosive mixture, an explosion usually takes place, the violence of which depends very largely on the speed at which the ignition is propagated. Bunsen found that in the case of a mixture of two volumes of hydrogen to one of oxygen the flame was propagated at the rate of 34 m. (37 yd.) per second, the velocity being much reduced by the presence of inert gases. With marsh-gas (CH_4) and air the greatest velocity was ·56 m. (22 in.) per second, and this was attained with a mixture of one volume of marsh-gas to eight and a half volumes of air, a mixture which contains less oxygen than is required for complete combustion. A flame with a velocity of about four and a half metres (4·9 yd). per second will pass through the wire-gauze ordinarily used for safety-lamps.

Explosion.—If an explosion takes place its violence, as remarked above, depends on the rate at which the flame is propagated. If it is in a closed vessel, vibrations may be set up which will enormously increase the speed of propagation of the flame, sometimes bringing it up to many hundred feet per second. Explosions of this kind are called by Wright "explosions of the second order", and it is to them that most of the damage done by explosions is due.

The velocities of explosion found by Dixon for mixtures with oxygen in the exact proportions for complete combustion were:

Hydrogen.		Acetylene.		Ethylene.		Methane.
2821	....	2391	....	2364	....	2322

metres per second

Dissociation.—Referring back to Deville's flame with a mixture of carbon monoxide and oxygen, the fact that the flame takes time to travel explains why the flame does not run back down the tube, but it does not explain why it spreads itself out into a flame of the ordinary form instead of at once igniting when it is released from the tube. This is due to dissociation.

The products in all ordinary cases of complete combustion are carbon dioxide and water, these being formed by the combustion of carbon, carbon monoxide, and hydrogen. If carbon dioxide and water be heated sufficiently strongly they are split up or dissociated into their constituents, water being broken up into hydrogen and oxygen, and carbon dioxide

into carbon monoxide and oxygen. It is quite evident that if hydrogen and oxygen, or carbon monoxide and oxygen, be brought together at a temperature higher than that at which this dissociation takes place, combination will be impossible, and therefore there can be no combustion.

If a mixture of hydrogen and oxygen be enclosed in a strong vessel and exploded, it is possible from a knowledge of the heat which will be evolved on combustion to calculate the pressure which the steam formed should exert. When the experiment is made it is always found that the pressure produced is less than the theoretical amount. The main reason for this is that combination is not instantaneous. As it progresses the temperature rises till the dissociation point is reached, when it can go no further, for this is the maximum temperature at which combustion is possible. As heat is lost by radiation, the gases cool and further combination takes place, and so on till combustion is complete. Another factor is the rather uncertain specific heat of steam at high temperatures.

Deville's experiment with the carbon monoxide and oxygen flame illustrates this very well. He carefully took the temperatures of all parts of the flame, and the results are recorded in the table.

Height above Burner.		Temperature.	Percentage of Gases.		
Mm.	Inches.		CO.	O.	CO_2.
67	2·64	Above melting-point of silver ..	·2	21·3	78·5
54	2·13	Melting-point of gold	6·2	28·1	65·7
44	1·73	Commencing white-heat of platinum	10	20	70
35	1·38	White-heat of platinum	17·3	24·8	57·9
28	1·10	Strong white-heat of platinum ..	19·4	26·5	54·1
18	·71	Intense white-heat of platinum ..	29	25·1	45·9
15	·599	Incipient fusion of platinum.. ..	40	32·9	27·1
12	·47	Melting-point of platinum	47	36	17
10	·39	Sparkling of melted platinum ..	55·3	35·3	9·4
0	..	.. ..	64·4	33·3	2·3

These figures at once explain the whole phenomena. As soon as combustion begins the temperature rises, and at the apex of the inner cone 10 mm. (·39 in.) above the burner it has reached the melting-point of platinum, which is above the dissociation point of carbon dioxide; so that no further combination is possible till the gases cool. This they do as they rise, and combustion again can take place, and this goes on till at the top of the flame all the carbon monoxide has disappeared and combustion is complete. From the very first the flame contains excess of oxygen, as some of the carbon monoxide is burned by the oxygen of the air. The length of the flame, therefore, is due to dissociation.

The dissociation temperature does not seem to be an absolutely fixed point, but varies with circumstances, it being in general raised by the presence of inert gases, and considerably lowered by contact with hot solids.

Dissociation plays a very important part in the practical applications of combustion.

Smoke.—Many hydrocarbon flames under certain conditions become smoky, the smoke being due to the separation of carbon under conditions which do not allow of its combustion. The cause of smoke is always imperfect combustion, due either to a deficient supply of air or to reduction of temperature. It is easy to see how the latter can be brought about. Air diffusing into a flame soon cools it; and if there be solid carbon in it this is likely to escape combustion. Smoke is always accompanied by other products of incomplete combustion.

Domestic Fire.—As an example of some of the causes which lead to smoke, an ordinary domestic fire may be considered. The whole question of the production and prevention of smoke will be discussed later.

Suppose, in the first instance, the fire-place to be full of glowing coke. As the air enters, combustion takes place and carbon dioxide is formed, $C + O_2 = CO_2$, together with some carbon monoxide either formed directly, $C + O = CO$, or by the reduction of carbon dioxide, $CO_2 + C = 2CO$. This coming into the air at the top of the fire burns with its characteristic blue flame, carbon dioxide being produced, $CO + O = CO_2$, and the combustion is complete. If now the fire be made up in the usual way, by throwing cold coal on the surface, all is changed. The reactions at the lower part of the fire go on as before, but the carbon monoxide in passing through the coal is cooled below the point at which ignition can take place. At the same time the heat below begins to act on the coal, and destructive distillation begins, gases, and tarry matter which forms a dense yellow smoke, being given off. These, being cool, do not ignite, but pass unburned to the chimney. After a time, as the heat penetrates, or perhaps when the fire is stirred, these gases ignite and burn with the bright flame characteristic of coal-gas. Smoke always indicates loss of fuel, not only because of the actual carbon which it contains, but also because the conditions which favour the production of smoke always favour the escape of combustible gases.

Heating by Contact or Radiation.—All fuels are burnt for heating purposes, and the methods of transferring the heat from the incandescent fuel to the object to be heated are of importance. Heat may be transferred in two ways—(1) by contact, as when a bar of iron is placed in a hot coke fire surrounded by the burning coke; (2) by radiation, as when an article is heated by being held in front of a fire. In many cases heating is necessarily by contact, as, for instance, in the blast-furnace, where the charge is heated by contact with the hot ascending gases, or with the hot fuel; and in others it is very largely by radiation, as when a room is heated by an ordinary house fire; and there are others in which both methods come into play.

Heating by contact of flame is not possible, except when the substance being heated is at a moderately high temperature. When flame is playing under a boiler it seems as if the heating were due to the actual contact

of the flame. This is not the case, as the flame cannot touch the comparatively cold surface—kept cold by the contact of the water—but is separated from it by a thin cold layer, across which heat can only travel by radiation. Gases are, as a rule, very bad radiators; hence the Bunsen burner, though very satisfactory for heating small articles with which the flame can come into contact, is a very poor source of heat for heating by radiation, and when it is so used, as in many gas fires, iron, asbestos, or other material is fixed so as to be heated by the flame and made to radiate, sometimes at the cost of hindering complete combustion. Water vapour is a very good radiator, and its presence no doubt materially increases the radiating power of many non-luminous flames. Carbon is one of the best radiators, and therefore the luminous flame with its separated incandescent carbon is much more efficient for heating by radiation than the non-luminous Bunsen flame which has been shown to have a radiation efficiency of about 10 to 15 per cent.

Amount of Air required for Combustion.—If we know the composition of a fuel it is an easy matter to calculate the amount of air which it requires for its complete combustion.

The air for all practical purposes may be taken as containing 21 per cent by volume and 23 per cent by weight of oxygen. When carbon burns to form carbon dioxide, 12 parts of carbon combine with 32 parts of oxygen to form 44 parts of carbon dioxide; so that 1 part of carbon will combine with 2·67 parts of oxygen to form 3·67 parts of carbon dioxide. If c be the percentage of carbon contained in a fuel which contains no other combustible material, then W, the weight of **oxygen** required for the combustion of 1 lb. of fuel will be given by

$$W = \frac{c \times 2{\cdot}67}{100} = (c \times {\cdot}0267) \text{ lb.}$$

The weight of air A will be

$$A = \frac{c \times 2{\cdot}67}{100} \times \frac{100}{23} = \frac{c \times 2{\cdot}67}{23} = (c \times {\cdot}116) \text{ lb.}$$

One part of hydrogen, when it burns, combines with 8 parts of oxygen to form 9 parts of water, so that the weight W of **oxygen** required for the combustion of 1 lb. of a fuel containing h per cent of hydrogen and no other combustible would be

$$W = \frac{h \times 8}{100} \text{ lb.,}$$

and A, the weight of air,

$$A = \frac{h \times 8}{100} \times \frac{100}{23} = \frac{h \times 8}{23} = (h \times {\cdot}348) \text{ lb.}$$

If the fuel contains c per cent of carbon and h per cent of hydrogen,

then W, the weight of oxygen required for the combustion of 1 lb. of the fuel, will be

$$W = \{c \times \cdot 0267 + h \times \cdot 08\} \text{ lb.}$$

and the **weight of air**

$$A = \{c \times \cdot 116 + h \times \cdot 348\} \text{ lb.}$$

If the fuel contains *o* per cent of oxygen, then *h* must be taken to stand not for the total but for the available hydrogen $(h - \frac{1}{8}o)$.

As 1 c. ft. of air at 32° F. and 30 in. of mercury pressure weighs ·0809 lb., the volume of air required for combustion is

$$V = \frac{c \times \cdot 116 + h \times \cdot 348}{\cdot 0809} \text{ (c. ft.).}$$

If the air be at any other temperature and pressure, this volume must be corrected.

If p is the actual pressure in inches of mercury, and t the temperature in degrees Fahrenheit, then by p. 135, the volume required is

$$v' = V \times \frac{30 \cdot 0}{p} \times \frac{460 + t}{492}.$$

The following formulæ are near enough for practical purposes. They are calculated for air containing an average amount of moisture.

c is the percentage of carbon in the fuel, and h the percentage of available hydrogen; A_w and A_v the weight and volume of air required, then

$$A_w = \cdot 12c + \cdot 35h;$$

and, taking 1 c. ft. of air at N.T.P. as weighing 0·0809 lb., the volume A_v would be

$$A_v = 1 \cdot 48c + 4 \cdot 34h \text{ (c. ft.).}$$

In practice excess of air must be used, so that the figures found as above must be multiplied by a factor. This will be, for gas furnaces about 1·5, for good grates about 2, and for defective grates 3 or more.

Products of Combustion.—The weight of the products of combustion will of course be the weight of the fuel consumed together with the weight of the air supplied; so that if W′ is the weight of the products of combustion, A_w is the weight of air, F the weight of fuel, and a the weight of the non-combustible portion or ash, then

$$W' = A_w + F - a.$$

The products of combustion will be carbon dioxide from the fuel, water partly formed by combustion of the hydrogen and partly moisture contained in the fuel, the nitrogen from the air and the excess of air; so that if c, h, o, and w be the percentages of carbon, hydrogen, oxygen, and

water contained in the fuel, and E the excess of air, the weight of the products of combustion will be

$$W' = \frac{c \times 3{\cdot}67 + h \times 9 + \{c \times 2{\cdot}67 + (h - \frac{1}{8}o)8\}\frac{77}{23} + w + E}{100}.$$

Heat carried away by Gases.—If it be required to know the heat carried away by the gases, this can be obtained by multiplying the weights of the products of combustion by their specific heats and the temperature at which they escape, less the standard temperature assumed. All that is required in practice is to know the amount of heat lost which could be usefully employed, and as heat below 212° F. would be of little value no note need be taken of the latent heat of steam.

The heat carried away will be

$$H = \frac{\{3{\cdot}67c \times {\cdot}2387 + (9h + w) \times {\cdot}4805 + \{2{\cdot}67c + 8(h - \frac{1}{8}o)\} \times \frac{77}{23} \times {\cdot}2485 + E \times {\cdot}2375\}}{100} \times t;$$

where t is the excess of the flue temperature above the standard temperature. If it be required to take into account the latent heat of steam, then $\frac{(9h + w) \times 966}{100}$ must be added.

A simpler and sufficiently accurate calculation is

$$H = (A_w + F - a) \times {\cdot}25 \times t.$$

Volume of Products of Combustion.—When carbon burns to carbon dioxide the carbon dioxide formed occupies the same bulk as the oxygen consumed ($C + O_2 = CO_2$). When carbon monoxide is formed the volume is twice that of the oxygen ($C + O = CO$). When hydrogen burns, the volume of the gas is two-thirds of that of the component gases or twice that of the oxygen ($2H + O = H_2O$). With gaseous fuels the reactions are more complex. Marsh-gas yields products which occupy the same volume as the gas burned and the oxygen used ($CH_4 + 2O_2 = CO_2 + 2H_2O$). With ethylene, C_2H_4, the products also occupy the same volume as the gas and oxygen, and with acetylene three-quarters of the volume. In general, therefore, for solid fuels the volume of the products of combustion may be taken as being equal to that of the air supplied, and with gaseous fuels as being equal to be sum of the volumes of the gas and the air.

CHAPTER V

The Laws of Change of Physical State

We have now to consider the processes of change of state; under what conditions those changes occur; and what quantities of heat are necessary to effect them.

Liquefaction.—Most solids can be melted by heat, although some substances melt only at very high temperatures, obtained by special means. Platinum and flint have been melted by placing them in crucibles of graphite in a furnace where a stream of air was maintained through very small pieces of coke, the combustion being thus very rapid, and giving the maximum temperature obtainable from burning carbon. Ruthenium, more refractory still, has been melted in the oxyhydrogen flame, and carbon itself by a combination of this with the electric arc. As our experimental resources have become more extensive, the number of substances that have never been melted has steadily fallen, and evidence is thus accumulating that all substances can exist in each of the three states—the solid, liquid, and gaseous—under certain particular conditions of temperature and pressure.

Phases.—The physical chemist speaks of each of the three states—solid, liquid, and gas—as a **phase.** Thus water in a bottle exists in two phases, the liquid phase and the gaseous phase above the liquid phase. We shall study first the passage from the solid to the liquid phase and vice versa.

Kinetic Theory of Heat.—Before we begin the experimental study we shall consider what kind of thermal effects may be expected to accompany a change of state. The physical state of a liquid is entirely different from that of a solid or a gas, and hence we should expect the passage from one phase to another to be accompanied by the liberation or absorption of definite amounts of internal energy. This liberation or absorption of energy will probably show itself as a thermal effect in the liberation or absorption of heat energy.

Laws of Fusion.—The laws of fusion may be conveniently demonstrated by placing small lumps of sulphur in a flask containing a thermometer, and applying heat. The mercury in the thermometer gradually rises till it indicates about 246° F., and remains stationary at that temperature till the whole of the sulphur is melted. When complete liquefaction has taken place, the temperature again rises. If the substance be then allowed to cool slowly, the temperature falls to 246° F., and remains stationary till the whole mass has solidified, when it again regularly falls. The distinguishing feature of the process is the fact that *although during the change of state heat enters or leaves the substance in large quantities, the temperature remains unchanged.*

The temperature at which the melting takes place is different for different substances, and is affected to some extent by the pressure to which the substance is subjected, when it melts.

When the pressure is that of the atmosphere, this temperature is called **the melting-point** of the solid.

If the above experiment be repeated with small lumps of sealing-wax or of glass, it will be found that the temperature never remains entirely stationary. The difference may be presented to the eye by a diagram such as fig. 17, where time is set off along the horizontal axis and temperature along the vertical axis.

Substances, then, may be divided into two classes according to their behaviour during liquefaction: in the one class the process is a gradual one, rise of temperature being accompanied by a gradual softening; in the other the process is abrupt.

When the process of melting takes place gradually as the temperature is raised, the substance cannot be said to have a melting-point. Such is the case with glass, sealing-wax, and wrought iron. The existence of this intermediate viscous stage is, however, of much practical use, since the substances when in this pasty condition are capable of being welded or drawn out into threads. Nearly all the substances which are of interest to the engineer, *thermodynamically*, such as water, gas, carbon dioxide, sulphur dioxide, ammonia, &c., have definite temperatures of change of state. Such substances only will now be considered.

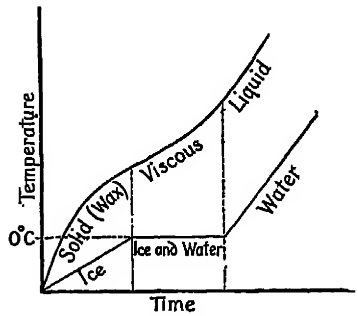

Fig. 17

Latent Heat.—The heat that enters a substance during a change of phase is said to be rendered **latent.** The word is a remnant of the language used at the period when heat was thought to be a subtle fluid, which in this process concealed itself from the thermometer.

In being absorbed by the substance, the latent heat has not been destroyed, but has done work upon the particles of the substance and communicated to them an equivalent amount of energy.

The latent heat of a substance is the number of units of heat which are absorbed (or emitted) when unit mass of the substance changes from one phase to another without change of temperature.

There are thus two latent heats to consider, one referring to the passage from the solid phase to the liquid phase, and the other to the passage from the liquid to the gaseous phase. The former is called the **latent heat of fusion,** the latter, the **latent heat of vaporization.** Both are *absorptions* of heat when the phase-change is from solid to liquid or from liquid to gas. Heat is evolved when the phase-change is in the opposite direction.

The **Laws of Fusion** are: (i) Under atmospheric pressure (30″ Hg) each substance begins to melt at a certain temperature, which is constant for the same substance.

(ii) The temperature of the solid remains at this fixed point (called the melting-point) during the whole time of the process of change.

(iii) The melting-point of a substance depends on the pressure to which it is subjected; if the substance expand on solidifying, the melting-point is lowered by increase of pressure; if the substance contract on solidifying, its melting-point is raised by increase of pressure.

(iv) Every substance during fusion absorbs heat. The quantity absorbed per unit mass is constant under the same conditions and is called the **latent heat of fusion.**

Determination of Latent Heat of Fusion.—The latent heat of a substance is usually determined calorimetrically by the method of mixtures (p. 141). Of substances ordinarily met with as liquids, the latent heat is determined by lowering the temperature of a known mass of the substance until it solidifies, placing the mass of solid in warm water, and noting the fall of temperature of the water as the substance liquefies. Details of the experiments will be found in any standard book on heat.

Numerical Values.—Tables of the latent heats of fusions of various substances will be found in Kaye and Laby's *Physical and Chemical Constants*. The figure for ice is 144 B.Th.U. at 32° F.

Solidification.—The process of solidification is the converse of liquefaction. It usually takes place when the liquid is exposed to a low temperature, but is difficult to bring about in the case of a few liquids. The study of solidification requires means for obtaining and measuring low temperatures.

Laws of Solidification.—(i) Under atmospheric pressure each substance begins to solidify at a certain temperature, which is always the same for the same substance.

(ii) The temperature of the liquid remains at this fixed point during the whole time occupied by the change. This fixed temperature, called the temperature of solidification, or in the case of water the freezing-point. is the same as the melting-point of the solid.

(iii) The temperature of solidification varies with the pressure.

(iv) Each substance during solidification gives out a quantity of heat equal to the heat rendered latent during fusion.

Change of Volume during Solidification.—It is a general rule that, during the operation of melting, expansion takes place, so that the liquid is less dense than the solid at the same temperature. In Kopp's experiments on this subject the substance was enclosed in a thermometer tube, and covered with some suitable liquid. The change of volume was deduced from the change of level of the liquid in the tube. The following results were obtained.

Substance.	Melting-point. Degrees F.	Volume at Melting-point.	
		(a) Solid.	(b) Liquid.
Phosphorus	111	1·017	1·052
Sulphur ..	239	1·096	1·150
Wax ..	147	1·161	1·166
Stearic acid	158	1·079	1·198

All these substances, as is usual, expand on melting.

In some cases, especially when solidification takes place slowly, the solid is deposited in the crystalline form, as in the case of bismuth; and when this occurs the peculiar molecular arrangement causes the solid to occupy more space than the liquid. Of such exceptional substances water is the most remarkable. To determine the density of ice Bunsen placed in a bulb a known mass of ice at 32° F. and filled the bulb up with mercury to a marked point. The ice was then melted, and while the water was still at 32° F. more mercury was passed into the bulb to fill it up to the mark. The mass of this additional mercury was found, and its volume was then known. This was the amount by which the ice contracted on melting. The value deduced for the density of ice was ·9167 gm./c. c. Nichols weighed ice in air and in oil of known density, and found the density of slowly frozen pond ice to be ·918 and of artificially prepared ice ·916. Vincent and Leduc obtained values practically the same. At −301° F., Dewar found the value ·93. At temperatures near 32° F. water is probably not homogeneous but consists of a solution of ice in water. If ice were heavier than water, so that it sank on forming, lakes and seas would gradually solidify from the bottom upward.

Cast iron is another important exception. This substance expands as it passes from the liquid to the plastic state, the amount of expansion being as much as 6 per cent; on further cooling to the solid condition it contracts to about the same extent. The fact of its expansion on solidification renders it peculiarly suitable for casting into moulds.

Bismuth appears to expand in passing from the liquid to the solid state by about 2·3 per cent; the specific gravity of the hot liquid is in fact greater than that of the cold solid.

This change of volume does not in all cases take place at the definite temperature known as the melting-point, but extends through a small range of temperature in the neighbourhood of that point.

Liquids below the Normal Temperature of Solidification.—Water may by special means be maintained, at atmospheric pressure, in the liquid state at a temperature considerably below 32° F. Fahrenheit did this by exposing to a low temperature water sealed up in a spherical vessel. Gay-Lussac maintained water at 10° F. by placing it in a flask, covering the surface with oil, and keeping it quite motionless. Despretz found that water in capillary tubes might remain liquid at −4° F. In all cases, however, sudden vibration or contact with air produced immediate

and rapid solidification; absence of air from the body of the liquid is necessary in order that the phenomenon may occur at all. Other substances may be made to give a similar result. Thus, if phosphorus be placed in water in a test tube, and the tube put in a beaker of water which is gradually warmed, the phosphorus becomes liquid above 111° F., remaining, however, at the bottom of the tube because of its greater density. If the beaker be then left to cool gradually, the phosphorus may remain liquid till the temperature is 86° F., but the immersion of a lump of ordinary (not amorphous) phosphorus at once provokes solidification.

The presence of a nucleus of some kind seems the necessary condition that solidification may take place at the normal temperature. Solidification of a liquid chilled below its point of congelation always takes place if a particle of the solid be dropped in the liquid. When the liquid thus commences to solidify, the latent heat of fusion is evolved, and the temperature immediately rises to the normal temperature of fusion.

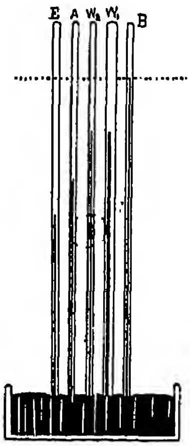

Fig. 18

We have now to consider the change of state from the liquid phase to the gaseous phase.

Formation of Vapours in Vacuo. — The term vapour is very generally applied to a gas which can be readily condensed, i.e. without resorting to extreme temperatures or pressures.

Let several barometer tubes be filled with mercury and stand inverted in a mercury trough (fig. 18). The mercury will stand at the same height in all the tubes, and in the first tube, B, may remain unchanged throughout the experiment.

By means of a curved pipette introduce a very small drop of water into the second and third tubes, W_1 and W_2, which are of different sectional areas. In each tube the drop of water will, on reaching the vacuum, rapidly evaporate, and the column of mercury will fall. If water continue to be passed into these tubes in very small quantities the mercury for a time continues to fall, but before long a level is reached below which the mercury ceases to fall. Any further water that may be introduced ceases to evaporate, and forms a liquid layer on the top of the mercury. The height of the column is the same in W_2 as in W_1.

Now let liquid alcohol be similarly passed up into tube A and ether into tube E. The result is of the same nature as before, but the amount of depression of the mercury column is different for each liquid.

We have supposed that the temperature throughout has remained at some ordinary value, e.g. about 60° F. Let now the flame of a spirit lamp be cautiously brought near the thin liquid layers above the mercury in the tubes, and it will be found that some of the liquid will evaporate and the columns will be more depressed. If, on the other hand, any

one of the tubes be cooled, some of the vapour condenses into liquid and the mercury rises.

The Laws of Vaporization in vacuo may be thus stated:

(i) A small quantity of liquid introduced into a vacuum rapidly evaporates at any temperature.

(ii) The quantity of liquid that can be made to evaporate *in vacuo* at a fixed temperature is limited, and is proportional to the volume of the vacuous space.

(iii) The vapour thus formed exerts a pressure on the containing vessel. The pressure for any particular substance at any fixed temperature (under ordinary circumstances and not in capillary tubes) does not rise beyond a certain value, which is called the **maximum pressure** of the vapour at that temperature.

(iv) Vapours differ from each other in the value of the maximum pressure they can exert at any temperature.

(v) The maximum pressure varies with the temperature, growing greater as the temperature rises and less as it falls.

Saturated and Unsaturated Vapours.—In the experiment just described the mercury continued at first to fall as more liquid was passed above it, but this fall ceased when a layer of liquid was formed above the mercury. Therefore in dealing with vapours in confined spaces an important distinction must be made. If liquid is present with its vapour, or, as it is commonly expressed, if the vapour is in contact with its liquid, that vapour is said to be **saturated**; it is exerting its maximum pressure, and is at its greatest possible density for that particular temperature.

If any additional pressure be put upon a saturated vapour it does not, as air does, increase in pressure as it diminishes in volume, but the vapour passes into the liquid state. If the pressure upon a saturated vapour be diminished, some liquid must evaporate in order that the vapour may remain saturated.

Generally speaking, if a vapour is not in contact with its liquid it is not saturated, although it is, of course, possible for saturated vapour to exist when not in contact with its liquid, as it does at the moment when on a slow rise of temperature the last particle of liquid evaporates.

If vapour be confined in a limited space with none of its liquid present, and in such quantity that it is exerting less than its maximum pressure, it is unsaturated or superheated. It can be converted into saturated vapour by sufficiently lowering its temperature or diminishing its volume.

Unsaturated Vapours.—Unsaturated or superheated vapours approximate to the condition of the more permanent gases. It has been established that vapours far removed from the conditions of condensation obey the gas laws, and their behaviour is in these respects not to be distinguished from that of the gases commonly called permanent.

Saturated Vapours.—If the vapour tubes stand in a reservoir sufficiently deep, the pressure on the vapour may be increased by lowering the

tubes in the mercury. If this be done slowly when the vapour is exerting its maximum pressure, it will be found that the height of the mercury column above the surface of that in the reservoir does not change. This shows that the pressure *of*, which is the same as the pressure *on*, the mercury does not change. The lowering of the tube is accompanied by a proportionate condensation of vapour into liquid. If a saturated vapour be compressed a portion liquefies, and if the changes of pressure and volume of the vapour take place slowly, so as to avoid disturbing effects, the pressure is constant and at its maximum value throughout. Whence it follows that Boyle's Law is completely inapplicable to saturated vapours.

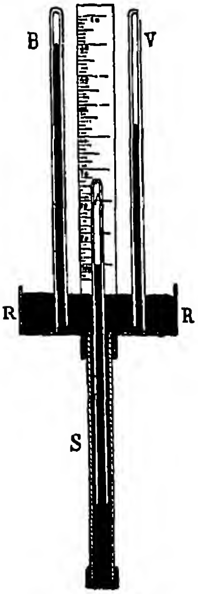

Fig. 19

Variation of Maximum Pressure with Temperature.—One of the most important investigations in this part of the subject is the relation between the maximum pressure and the temperature. The classical experiments on steam were made by Regnault (1810–78). His observations extended from a point well below the freezing-point to about 450° F. His experiments showed that corresponding to every temperature there is a definite maximum vapour pressure. This question will be taken up in further detail in the next chapter.

Mixture of Gas and Vapour.—Thus far we have spoken of single liquids evaporating *in vacuo*. Dalton examined the process when a liquid evaporated into a space already occupied by air, gas, or vapour by means of an apparatus similar to that shown in fig. 19. In that figure RR represents an iron reservoir terminating at the bottom in a vertical iron tube S, the whole containing mercury. In the bath stand three barometer tubes. Of these B serves as a standard barometer with which to compare the others, and the mercury in the tube V also originally stands at the same height as that in B with a vacuum above the mercury.

Now let there be passed into the tube V a sufficient quantity of some volatile liquid to saturate the space; this will evaporate, exert pressure, and depress the mercury column through x mm., which is the measure of the pressure of the saturated vapour *in vacuo* at the temperature of the apparatus.

The tube A when placed in position contains some dry air. Its reading is therefore y mm. below that of B, and the position of the mercury surface is marked on the tube. The same liquid as was used for the tube V is then passed into A in sufficient quantity to saturate the space above.

The total depression of the mercury column is now less than $x + y$ mm., because the air having expanded to fill the larger space exerts a pressure of less than y mm., its former value. To eliminate this effect the tube A is lowered into the reservoir until the surface of the mercury

column stands again at the point marked; when this is done the air occupies the same volume as at first and therefore exerts the same pressure.

The total pressure of the gases in A is measured by the difference of level between the mercury surfaces in A and B, and this is found to be exactly $x + y$ mm., of which y mm. is known to be due to the confined air, and therefore x mm. must be due to the vapour.

This value being the same as that for the vapour in the tube V, it appears that *the quantity of vapour which can exist in any limited space depends on the temperature only, and is the same whether that space be vacuous or already occupied by gas.* This is Dalton's law. It is assumed, of course, that the substances do not act chemically on each other.

The *rate* of evaporation is much greater *in vacuo* than when air or another gas is present.

If liquids be mixed, it is found that if the liquids dissolve each other the vapour pressure of the mixed liquid is less than the sum of the pressures of the constituents, but if the liquids are merely in juxtaposition without real intermixture the vapour pressure is equal to the sum of the pressures of the constituent vapours.

Density of Moist Air.—The mass of water vapour present in a given volume of damp air has often to be calculated. Proceeding on the assumption that the vapour obeys the gaseous laws, the expression may be obtained for any given temperature and pressure by the method shown on p. 168, remembering that 0·641 is the density of aqueous vapour compared with that of air. Thus we have:

Pressure in Inches of Mercury.	Temperature.	Mass in Pounds of one Cubic Foot.
30	32° F. (460)	$\cdot 641 \times 0{\cdot}0809$
p	t° F.	$\cdot 641 \times 0{\cdot}0809 \times \frac{p}{30} \times \frac{492}{\theta}$
	$\theta = 460 + t$	

θ and 492 being the absolute temperatures corresponding to t° F. and 32° F. respectively.

Then in a litre of moist air at the absolute temperature θ and at a total pressure P, the air containing vapour at a pressure p, we have

(i) A cubic foot of water vapour at a pressure p, weighing

$$\cdot 641 \times 0{\cdot}0809 \times \frac{p}{30} \times \frac{492}{\theta} = 0{\cdot}850 \times \frac{p}{\theta} \text{ lb.}$$

(ii) A cubic foot of dry air at a pressure $P - p$, weighing

$$0{\cdot}0809 \times \frac{P - p}{30} \times \frac{492}{\theta} = 1{\cdot}326 \times \frac{P - p}{\theta} \text{ lb.}$$

When a gas is collected over water, evaporation takes place at each

bubble and the collected gas is generally saturated. The quantity of dry gas may be ascertained from the above expression.

Ebullition.—When a certain temperature is reached in heating a liquid, the process of boiling begins, a process that for any fixed pressure takes place at one definite temperature only, when bubbles of gas are formed within the mass of the liquid. The process of ebullition can be readily observed by boiling water in a flask over a spirit lamp. At first small bubbles are seen to form on the sides of the glass and to rise to the surface; these consist of air. Then at certain points of the heated lower surface larger bubbles are formed which diminish and collapse when they pass into the colder water above; these are steam. When these steam bubbles cease to collapse but reach the surface and pass into the space above, the water boils. If a thermometer be placed in the water it will continue to indicate a rise of temperature until this third stage is reached, when the mercury will remain stationary although heat continues to be absorbed by the liquid until the water has all boiled away.

If the process be repeated with alcohol, ether, sulphuric acid, and other liquids, it will be found that each liquid boils at a different temperature.

Effect of Pressure on the Boiling-point.—When we speak of **the boiling-point** of a liquid we mean the temperature at which the liquid boils under normal atmospheric pressure, for a liquid can be made to boil at temperatures considerably above or below that at which boiling usually takes place. In fact, by simply immersing the same thermometer on different days in boiling water, perceptible differences in temperature may be found, due to the ordinary variations in atmospheric pressure.

In an experiment devised by Franklin a flask is half-filled with water, and the water is made to boil for some time till most of the air has been expelled from the flask, which is then corked and inverted. If a lump of ice be then placed on the flask, ebullition recommences and continues until the temperature of the water is far below 212° F. This result is due to the condensation of the steam above the water, and the consequent diminution of the pressure on the surface.

Any liquid boils at that temperature at which the pressure exerted by its vapour is equal to the pressure upon the surface of the liquid.

Laws of Boiling:

(i) At any fixed pressure each liquid begins to boil at a certain definite temperature. If the pressure be normal (30″ Hg), the temperature of the vapour during ebullition is called **the boiling-point.**

(ii) During the process of boiling this temperature remains unchanged.

(iii) The temperature at which any liquid boils varies with the pressure.

(iv) During the process heat is absorbed or rendered latent.

Retardation of Boiling.—The process of boiling has often some difficulty in starting. This appears to be due to the absence of air. When water commences to boil the bubbles formed on the hot surface are small, and increase in size as they rise through the liquid. The origin

of these bubbles is a small quantity of air present at their points of formation, into which the water *evaporates*, and unless there be present within the liquid small bubbles of air into which this evaporation can take place, boiling does not commence at the temperature corresponding to the pressure. Thus pure water under the normal pressure boils at 212° F. in a metallic vessel, but in a clean glass vessel its temperature may be raised several degrees higher before boiling commences.

Donny placed some water in a bent glass tube which had been most carefully cleaned, boiled the water for a long time to expel all air, and sealed the tube during ebullition. The water thus confined in a space almost perfectly destitute of air was raised to 280° F. without boiling, but at that temperature ebullition commenced with explosive violence.

When a liquid is in this **superheated** condition the introduction of the smallest quantity of gas or vapour produces violent ebullition. This effect follows the introduction of a solid rod, but it is almost certainly due to the air entrained on the surface of the rod. The gas need not be air, however, for the sudden ebullition takes place when a small quantity of any superheated liquid is decomposed by the passage of an electric current between two terminals immersed in the liquid, if one of the products of decomposition be a gas.

Latent Heat of Vaporization.—When pure water boils away into steam a large amount of heat is added to the latter, but the steam is no hotter than the boiling water. Every liquid which does not change its composition during the process absorbs heat in passing into the gaseous state. This heat has been transformed into molecular energy. Conversely, when a vapour is converted into a liquid, there is a reduction in the molecular energy with the appearance of heat.

The method of finding the latent heat of vaporization is to evaporate the liquid in a vessel, and then to allow the vapour to pass through a tube to a reservoir or spiral immersed in cold water, where it condenses into a liquid, and in so doing gives out its latent heat to the water with which it is surrounded, and whose rise in temperature allows the quantity to be determined. This is substantially the method of mixtures. Details of recent experiments will be found in any standard book on physics.

Total Heat of Steam.—It is shown above that the heat measured within the calorimeter is the result of two processes—the condensation of the steam and the cooling of the water thus formed. The total heat given out by unit mass of steam at t° F. in condensing and cooling to 32° F., or what is the same thing, the heat required to raise unit mass of water from 32° F. to t° F., and then to evaporate it at that temperature, Regnault called the **total heat of steam at t° F.***

The values of the total heats for different temperatures are given in Steam Tables.

Comparison of Latent Heats.—The following method has been employed for the comparison of latent heats of evaporation. The liquid

* See p. 182 for the modern definition of Total Heat.

is placed in a glass bulb which is enclosed in and communicates with a surrounding vapour jacket. Inside the bulb is a spiral of fine platinum wire ending in thick terminals fused through the glass. An electric current passed through the wire develops heat, and the vapour passes round the jacket. When the liquid has been boiling for some time and the whole has reached a steady temperature, then all the heat developed in the wire is spent in converting liquid into vapour and none in raising or maintaining the temperature. The energy supplied is capable of very exact determination.

For comparison, two such bulbs containing different liquids are arranged in series, and the ratio of their losses in weight is inversely as their latent heats of evaporation.

Employing this method, Ramsay and Marshall obtained the following results:

Substance.	Latent Heat, L. (B.Th.U.)	Boiling-point, $t°$ F.	Molecular Weight, m.	Value of $m \times L \div \theta$.
Benzene	170	176	78	20·8
Toluene	156	231	92	20·7
Alcohol (ethyl) ..	390	173	46	28·3
Water	966	212	18	25·8
Methyl butyrate ..	143	217	102·1	21·5
Mercury	122	674	200·6	21·5

Other liquids gave a result for column five of about 21. From many such observations Trouton has been led to formulate the law that *in liquids the molecular latent heat of vaporization* ($m \times L$) *divided by the absolute temperature of the boiling-point is a constant.*

The method indicated above has been applied to liquid oxygen, the liquid being contained in a vacuum vessel that stood in liquid air. The value of L obtained for oxygen was 104·5.

CHAPTER VI

Tables of the Properties of Vapours

The vapours in which engineers are specially interested are those employed in the steam-engine and in refrigerating machines. These vapours are steam, carbon dioxide, sulphur dioxide, ammonia, and ethyl chloride. The properties of these vapours, especially those of steam, have been studied in detail by physicists and engineers, and the results of the experiments are summarized in books of tables of the properties of vapours. We shall deal first with steam.

What a Vapour is.—A vapour is a substance in the gaseous phase.

A pure vapour therefore consists of a substance in one phase only. Vapours differ physically from the so-called permanent gases in the facts that:

1. Under ordinary conditions of temperature and pressure, two phases of the substance, the liquid and the gaseous, can exist together, as, for instance, in wet steam. In the permanent gases the two phases can only exist together at excessively low temperatures, i.e. under quite abnormal conditions of temperature.

2. Their characteristic equations are usually much more complicated than those for permanent gases.

It will be convenient to trace out the conversion of one pound of water into one pound of steam *under constant pressure* in a cylinder.

Change of State under Constant Pressure.—We will suppose a pound of water at 32° F. to be contained in a long, upright cylinder, the piston of which is equivalent to a load of P lb. per square *foot*. This piston rests in contact with the water at the bottom of the cylinder, and the water is at a pressure of P lb. per square foot everywhere. We now apply heat to the bottom of the cylinder. The water warms up until it reaches a temperature, T, such that the maximum vapour pressure corresponding to this temperature is equal to the applied pressure P. The water absorbs the so-called *water-heat* in this process. It now begins to evaporate and the evaporation continues, so long as heat is supplied, until all of it is turned into steam. There is no increase in temperature during evaporation. During evaporation, an amount of heat L, *the latent heat of steam*, is absorbed. The heat input is therefore $h + L$, where h is the water-heat and L the latent heat. This heat is called the *heat of formation*.

When evaporation is complete, we have 1 lb. of steam at temperature T and pressure P, and it is *saturated* vapour exerting its maximum vapour pressure. There is no reason, however, why we should stop adding heat. If we go on adding heat, the vapour expands and pushes the piston out against the external pressure, and gets hotter than the temperature of saturated steam corresponding to the pressure P. The steam is now *unsaturated*, since it is not as dense as it can be at its new temperature t. Steam in this condition is called *superheated steam*. If t is the actual temperature of the steam, the superheat is $S(t - T)$, where S is *the specific heat of steam*. This quantity, **the specific heat of steam**, has been very carefully measured. The heats we have so far discussed are:

1. The water-heat, h.
2. The latent heat, L.
3. The superheat, $S(t - T)$.

Mechanical Features of Formation of Steam.—We must now consider another aspect of the formation of the steam. The piston has obviously been pushed out in the process of evaporation, as the volume of a pound of steam is much greater than the volume of a pound of water.

This means that mechanical work has been done. It is well known that when a piston is pushed out in a cylinder against a pressure P, the mechanical work done, W, is equal to the pressure P, multiplied by the increase in volume ΔV, i.e. $W = P\Delta V$. We have seen, p. 123, that heat is simply a form of mechanical energy. Consequently, since, in forming the steam, we have done a certain amount of external mechanical work, some of the heat of formation must have been used in doing this mechanical work. The water-heat we actually supply in raising the temperature of the water from 32° F. to T° F. is h, and during this change of temperature the mechanical work done is insignificant, as there is no appreciable difference in volume between water at 32° F. and T° F. Consequently all the heat supplied must go in increasing the internal energy of the water. The conversion of the water into steam is accompanied by a considerable amount of expansion. If V is the volume of one pound of saturated steam, and ω is the volume of one pound of water, the expansion is $V - \omega$, and the work done is $P(V - \omega)$. The heat equivalent of this is

$$\frac{P(V - \omega)}{J} \text{ B.Th.U.,}$$

where J is Joule's mechanical equivalent of heat, see p. 189. This amount of the latent heat must therefore be accounted for by the external work done in the expansion, leaving, to increase the *internal* energy of the pound of steam itself,

$$\left[L - \frac{P(V - \omega)}{J}\right] \text{ B.Th.U.}$$

The total increase in the internal energy of the pound of material in question is given by

$$E_T - E_{32} = h + L - \frac{P(V - \omega)}{J},$$

where E_{32} is the internal energy of water at 32° F. and a pressure P lb. per square foot.

The quantity $(E_T - E_{32})$ is called *the internal energy* of saturated steam at temperature T, and is written shortly E. We thus get the equation

$$E = h + L - \frac{PV}{J} + \frac{P\omega}{J}$$

$$= h + L + \frac{P\omega}{J} - \frac{PV}{J};$$

$$\text{i.e. } E + \frac{PV}{J} = h + L + \frac{P\omega}{J}.$$

This quantity $\left(E + \frac{PV}{J}\right)$ is called by Callendar the **total heat** of saturated steam at temperature T.

It differs from the *heat of formation* $(h + L)$ only by $\frac{P\omega}{J}$, which is a very small term. For instance at atmospheric pressure and 212° F., $\left(E + \frac{PV}{J}\right)$ is 1150·74 B.Th.U., and $\frac{P\omega}{J}$ is given by

$$\frac{14\cdot7 \times 144 \times 0\cdot016}{778} = 0\cdot0435 \text{ B.Th.U.},$$

hence, for practical purposes, either the *heat of formation* or *the total heat* is 1150 B.Th.U.

This quantity H, the total heat, is a very important one in thermodynamics. For a change of state *at constant pressure* we have

$$\begin{aligned}
\text{Increase in H} &= \text{increase in E} + \frac{P}{J} \times \text{increase in V}, \\
\text{or } \Delta H &= \Delta E + \frac{P\Delta V}{J}, \\
\text{i.e. } (H_T - H_{32}) &= (E_T - E_{32}) + \frac{P}{J}(V - \omega), \\
&= h + L, \text{ exactly}, \\
\text{i.e. the change in total heat} &= \text{heat of formation at constant pressure} \\
&= \text{heat supplied in the given change of state.}
\end{aligned}$$

Steam Tables.—The properties of steam may therefore be conveniently studied by a series of experiments made at different constant pressures. These experiments have been made and the results are tabulated in books of *steam tables*. The original experiments were made by Regnault, but his results have been found to be slightly inconsistent among themselves. Professor Callendar, who has studied the question in great detail, has found that the relationship between the temperature, pressure, and volume of one pound of dry saturated steam or of one pound of superheated steam can be represented by a single equation which is closely in line with the best experimental determinations. The equation is

$$V - 0\cdot01602 = \frac{0\cdot5948T}{P} - 0\cdot4213\left(\frac{671\cdot58}{T}\right)^{\frac{10}{3}}.$$

In this formula V is in cubic feet, P in pounds per square inch, and T in degrees Fahrenheit absolute.

Working from this equation Callendar deduces by the principles of thermodynamics the different important heat quantities, and the quantities so calculated have the advantage over Regnault's figures that they are logically consistent one with the other, whereas some of Regnault's are not. He has published* a set of steam tables for engineer's use of which a selection is given (pp. 225–9).

* *The Callendar Steam Tables*, Arnold, 1915.

The important figures tabulated in a book of steam tables are the following ones:

1. The specific volume of steam, that is, the volume in cubic feet of 1 lb. of saturated steam.
2. The temperature of the steam.
3. The total heat of the steam.
4. The entropy of 1 lb. of saturated steam.

The "entropy" is a thermodynamic quantity, which will be described in the chapter on thermodynamics. It is of great importance in calculating the performances of heat-engines and refrigerating machines. The figures are usually given in terms of the pressure and refer to saturated steam.

In engineering practice superheated steam is largely used and tables are also provided giving the total heat and entropy of steam, superheated to different temperatures at given pressures. These tables take account of the variations in the specific heat of steam.*

Wet Steam.—In most of the engineering calculations concerning engines and turbines, the steam is not dry but wet, that is to say, the working substance is present in the two phases, the liquid and the gaseous phase. It is usual to deal with the calculations of wet steam by employing a factor which is called the *dryness fraction*. If q is the dryness fraction, then q lb. is present as vapour and $(1 - q)$ lb. as water, per pound of material, so that:

1. The latent heat of one pound of wet steam is qL.
2. The total heat of one pound of wet steam is $H_\omega + qL$, where H_ω stands for the total heat of one pound of water at the temperature of the steam.
3. The volume of one pound of wet steam is $qV + (1 - q)\omega$, which is very nearly equal to qV unless the steam is very wet.

Vapours used in Refrigerating Machines.—The vapours used in refrigerating engineering are ammonia, carbon dioxide, sulphur dioxide, and ethyl chloride. The last-named vapour has been used comparatively recently. The refrigerating engineer requires the same kind of thermal information concerning these vapours as the steam engineer needs concerning steam, i.e. the temperature, specific volume, total heat, and entropy of the saturated and superheated vapours in terms of pressure. The principal tables available are the following:

Ammonia.—(1) *Thermodynamic Properties of Ammonia*, by F. G. Keyes and R. B. Brownlee (John Wylie and Sons). (2) *The Properties of Saturated and Superheated Ammonia Vapour*, by G. A. Goodenough and W. E. Mosher (University of Illinois).

Carbon dioxide.—(1) "The Properties of Saturated Vapour of CO_2", by G. C. Hodsdon (*Ice and Cold Storage*, November, 1912). (2) Two papers on "The Thermal Properties of Carbonic Acid at Low Temperatures", by C. F. Jenkin and D. R. Pye (*Philosophical Transactions of the Royal Society of London*, Vol. 213, 1913, p. 67, and Vol. 215, 1915, p. 353).

* *Steam Tables and Diagrams*, by L. S. Marks and H. N. Davis (Longmans).

Ethyl Chloride.—"The Properties of Ethyl Chloride, C_2H_5Cl", by G. C. Hodsdon (*Ice and Cold Storage*, December, 1919).

A mine of information concerning these vapours and the best methods of calculating the performances of refrigerating machines is published in a report by the Refrigeration Research Committee of the Institution of Mechanical Engineers, published in 1914. By the courtesy of the Institution, the publishers are able to reproduce Charts I, II, and III from this Report (see pocket, end of volume).

CHAPTER VII

Thermodynamics

The science of thermodynamics is the study of the relations between heat and mechanical energy, and of the conversion of either into the other.

Energy.—*Energy is working capacity or the power of doing work*, and, like work, is measured, scientifically, in ergs or foot-poundals. Engineers measure mechanical energy in foot-pounds. A foot-pound is equal to 32·2 foot-poundals (i.e. g foot-pdls.).

It requires an expenditure of work to set a mass of matter in motion, and the mass in coming to rest does an amount of work in some form or other exactly equal to that spent upon it.

Hence while the body was in motion it possessed the power of doing this work—i.e. it possessed energy. All moving bodies are by virtue of their motion endowed with energy, which, when it exists in this particular mode, is called **kinetic energy.**

If a mass is raised through a vertical height a certain amount of work is done upon the mass, and while it remains in its raised position it possesses an equivalent amount of energy. This energy of position is called **potential energy.** In all cases, if no energy has been dissipated, the work done on a body is the measure of the energy communicated to it, whether the work has been spent in actually producing motion (kinetic energy) or in moving the body against the action of force into such a position that it is capable of doing work (potential energy).

When a body is in the act of falling, its energy is changing from potential to kinetic. At the instant of reaching the ground all its potential energy has been converted into kinetic, and on striking the ground the kinetic energy of the moving mass is converted into heat—another form of energy. During the fall the total quantity of energy (kinetic + potential) remains constant; one increases just as much as the other decreases. A vibrating pendulum is an example of continual interchange of kinetic and potential energy.

All physical operations consist in the transference of energy from one body to another.

Kinetic Energy of a Moving Body.—The kinetic energy of a moving particle of mass m lb. is

$$\frac{1}{2}\frac{m}{g}v^2 \text{ ft.-lb.},$$

where v is the speed in feet per second of the particle, and g the acceleration due to gravity, i.e. 32·2 ft. per second per second.

Power.—Power is the *rate at which a machine or agent does work. It is measured by the number of units of work done in unit time.*

A **horse-power** is 550 ft.-lb. per second.
A **watt** is 10^7 ergs per second; 1 watt $= \frac{1}{746}$ h.p.

General Principles of the Science of Energy.—The principles of work and energy were originally derived from the study of machines. Practical mechanics, which preceded theoretical, was directed to the object of getting work out of machines. For many years much effort was expended in the attempt to discover perpetual motion, i.e. to find a machine that should work of itself, that should give out energy when none was supplied to it. All attempts failed, and these points gradually became clear—that a machine to which no energy is supplied can do no work; that a machine may multiply force but cannot multiply energy; and that the true function of a machine is to *transmit* energy, not to create it.

Mechanical energy is measured by the product of force by distance; and in proportion as a machine multiplies the force applied to it, it divides the distance through which the point of application of the force moves. Thus, if by a pulley a force equal to the weight of 1 lb. raises a mass of 8 lb. against the force of gravity, then the force must work through 8 ft. in order that the mass may be raised through 1 ft.

In driving a machine at least as much work must be done on it as can be obtained from it. Mechanical energy cannot be created. This is one aspect of the principle of the conservation of energy. All the simple machines are conservative, and therefore all ideal complex machines are conservative also.

It was perceived, however, that in all machines there was an apparent *loss* of energy; no machine gives out as much work as is done upon it. This energy was for a long time supposed to be lost, but it was at last noticed that the *disappearance of mechanical energy is usually accompanied by the production of heat.* If, then, heat were a form of energy, the principle of the indestructibility of energy would be made more probable. The experiments of Rumford and Davy (p. 130) led independently to the suggestion that heat was molecular motion, and therefore a form of energy. The investigations of Joule turned the suggestion into a certainty by proving that *the heat developed when mechanical energy is destroyed is always exactly proportional to the energy that thus disappears.* Friction is the chief mode of converting mechanical energy into heat. To these considerations must be added the fact of the existence of heat engines such as the locomotive,

the mechanical motion of which is wholly dependent upon the heat produced in the furnace. Heat thus is proved to be a form of energy. But there are other forms also, and it is necessary here to allude to some of them. When a gun is fired a large quantity of energy is communicated to the shot, consequent upon the chemical actions that take place between the ingredients of the powder. As we have seen in Chapter I, chemical action in general leads to a production of heat, i.e. of energy.

If electrical currents be started in coils of wire suitably placed near each other, motion of the coils ensues, the energy of their motion being taken from that of the electric current. The coils also become heated by the passage of the current. An electrical current is therefore a form of energy. When a voltaic cell is used to produce the current, the origin of the energy is the chemical actions going on within the cell.

Without attempting a complete enumeration of the forms of energy we may note the following as the chief: mechanical energy, electrical energy, the energy of gravitation, heat, radiant energy, the energy of chemical affinity.

The science of energy embraces three great principles:

The Transmutation of Energy.
The Conservation of Energy.
The Degradation (or Dissipation) of Energy.

(i) *Transmutation of Energy.*

The first of these principles is that there are several forms of energy, and that energy may be converted from one form to another. From experiments such as those alluded to above, it appears that energy is readily capable of transmutation from the mechanical and electrical forms into the thermal form. In fact every kind of energy may be converted into heat. Other transmutations do not take place so readily as this. Many illustrations of transmutations of energy might be given. Thus the energy of a waterfall is sometimes employed to give motion to the armatures of dynamo machines by which it is converted into the energy of electrical currents, that may give out their energy either in the mechanical or thermal form many miles away. The energy of the tides may be thus utilized. Twice a day the sea rises round our coast, and if in falling it were made to turn water-wheels, a vast amount of work could be obtained.

A man may be regarded as a machine for transmuting the chemical energy contained in his food into mechanical work. He may work a treadmill which may turn the armature of a dynamo machine and thus produce an electric current. It might be arranged that this electric current in passing through a wire should heat the wire, in which case the energy would have passed from the mechanical into the electrical, and from the electrical into the thermal form. In any case the chemical energy contained in his food maintains the heat of his body.

(ii) *Conservation of Energy.*

In these transmutations of energy none is destroyed. *Energy is alike indestructible and uncreatable* by any natural agency. This principle is not capable of simple direct proof, but "it is the one generalized statement that is found to be consistent with fact in all physical science" (Clerk-Maxwell). It was suggested by purely mechanical considerations, and was made more probable by the fact that when there was an apparent loss of energy in any machine heat was produced. The establishment of a numerical relation between the units of heat and of work goes far towards the establishment of the general principle, and many theoretical results deduced from it have been confirmed by observation.

(iii) *Degradation or Dissipation of Energy.*

The different forms in which energy can exist may be arranged in the order of their availability to us. Those at the top of such a list are called the higher forms, and those at the bottom the lower forms. Mechanical energy is one of the highest forms, and the heat contained in a body at low temperature is one of the lowest. Heat contained in a body at a high temperature is more available than heat contained in a body at a low temperature. When energy passes from a more available to a less available form, it is said to be degraded.

Such transmutations of energy as those mentioned above cannot continue indefinitely. At each change the available energy grows less, because at each step of the process some of the energy runs down into the lower form of diffused heat. Thus, for example, we cannot convert all the energy of a waterfall into mechanical work, for the water running away possesses kinetic energy, which is gradually dissipated, i.e. it takes a form which we cannot employ for any useful purpose. We can always transform mechanical energy into heat, but this heat cannot be all transmuted back again into mechanical energy.

The chief means by which energy of motion is dissipated is friction. Energy expended in friction generally becomes diffused, and, except by special means, cannot be recovered.

Since all our physical operations consist in transformations of energy, it would appear that when all the energy in the universe has run down to the lowest form all such operations must cease—unless some important factor is not at present known. The law of dissipation of energy may be thus stated: *Any transformation of energy is accompanied by a degradation of energy.*

First Law of Thermodynamics.—We have already referred to the fact that heat is a form of energy, and that 1 B.Th.U. is equivalent to 778 ft.-lb. of energy. This fact gives us a relation between the unit of mechanical energy and the unit of heat, i.e. of heat energy. "*When work is transformed into heat or heat into work the quantity of work is mechanically equivalent to the quantity of heat*" (Clerk-Maxwell). Symbolically expressed

$$W = JH. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

If H is in B.Th.U. and W in foot-pounds, J is 778. This fact is often called the *first law of thermodynamics*. The law follows at once from the kinetic theory of heat.

Heat is supposed to be due to the motion of the molecules of a body. If a body A is hotter than one B, the molecules of A are more violently agitated than those of B. If we take a pound of material, we have a definite number of molecules. If we raise the temperature of it 1° F., we impart a greater state of agitation to these molecules and hence we must increase the mechanical energy of each by a definite amount. We therefore increase the mechanical energy of the sum total of the molecules by a definite amount. This amount of mechanical energy is 778 ft.-lb., by experiment. *We can therefore measure both mechanical and thermal energy in heat units*, if we divide the mechanical measure by 778, thus 100,000 ft.-lb. are equivalent to

$$\frac{100,000}{778} = 128 \cdot 5 \text{ B.Th.U.}$$

Henceforth we shall suppose all energy to be measured in units of heat.

The Mechanical Equivalent of Heat: Recent Determinations.—Many careful evaluations of the mechanical equivalent of heat have been made.

Griffiths employed the electrical method. The calorimeter containing the wire and the water to be heated was closed by an air-tight lid, and was suspended in an enclosure which had double walls, the space between them being filled with mercury. The formula $JH = C^2rt$ (p. 125) may be written $JH = \frac{E^2t}{r}$ where E is the difference of electrical potential between the terminals of the wire whose resistance is r and t is the duration of the test. Griffiths kept E constant by means of a regulating rheostat, and measured E by balancing it against a number of Clark standard cells in series. The mercury thermometer employed was standardized in terms of the hydrogen scale. The temperature range was between 59° F. and 77° F.

Messrs. Schuster and Gannon also employed the electrical method, writing the equation in the form $JH = CEt$. The value of the current C was ascertained from the quantity of silver deposited in a silver voltameter, and E was measured by balancing the e.m.f against 20 Clark cells in series.

The experiments of Callendar and Barnes furnish one of the most accurate measurements of the value of J.

Employing the friction method, Messrs. Reynolds and Moorby made some experiments on a large scale for the determination of the mean value of J between 32° F. and 212° F. They employed a steam engine of 100 h.p. fitted with a hydraulic brake which consisted of paddles working in a vessel of water. The water flowed into the vessel at about 32° F. and out at about 212° F.

Summary of Results.—The values obtained for J by the various experimenters and methods are:

	Method.	Temperature. Degrees F.	Foot-pounds per 1 Degree F.
Joule (final)	Friction	60	772·6
Rowland	"	41 to 95	777·6
Griffiths	Electrical	59 to 77	779
Schuster and Gannon ..	"	—	779
Reynolds and Moorby ..	Friction	32 to 212	777
Callendar and Barnes ..	Electrical	—	777·6
Barnes	"	41 to 95	777
Miculescu	Friction	50 to 55	776·6
Cremieu and Rispail ..	—	—	777·6
Mean of modern values..	—	—	778

It is now generally held that Joule's value is too low; the discrepancy being probably due to errors in the thermometers he used.

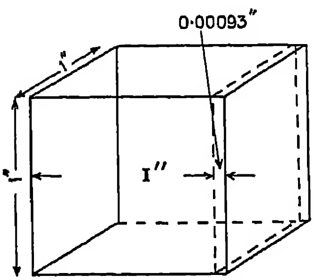

Fig. 20

Internal and External Work.—Suppose we have a cube of copper (fig. 20), each side of which is 1 in. long at 32° F., and suppose we heat it to 132° F. The coefficient of expansion of copper is $9·3 \times 10^{-6}$, therefore the length of each side at 132° F. is

$$L_{132} = 1(1 + 9·3 \times 10^{-6} \times 100)$$
$$= 1·00093 \text{ in.}$$

The area of each face is approximately 1 sq. in. Suppose the cube expands against atmospheric pressure, then the force resisting expansion is $p \times$ area of face $= (p \times 1)$ lb., where p is the atmospheric pressure in pounds per square inch. The work done in the displacement of one pair of opposite sides is

$$(p \times 0·00093) \text{ lb-in.}$$

But each pair of opposite faces does this amount of work, hence the total work done is

$$(3p \times 0·00093) \text{ lb.-in.}$$

Now $3 \times 0·00093$ is the increase in *volume* of the cube nearly, hence if we put δv for the increase in volume,

$p\delta v$ is the work done in expansion.

If p is in pounds per square *foot*, and v in cubic feet, $p\delta v$ is in foot-pounds.

Dividing this quantity by J we get the heat equivalent of this work,

$$\frac{p\delta v}{J}.$$

If we call H_e the heat equivalent of this *external* work done, we get

$$H_e = \frac{p\delta v}{J}. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

In the example we are taking, p is 14·7 lb. **per square inch,** and δv is $3 \times 0{\cdot}00093$ c. in.

$$\therefore p\delta v = \frac{(14{\cdot}7 \times 144) \text{ lb. per } \textit{square foot} \times \left(\dfrac{3 \times 0{\cdot}00093}{1728}\right) \text{ c. ft.}}{778}$$

$$= \frac{14{\cdot}7 \times 144 \times 3 \times 0{\cdot}00093}{778 \times 1728}$$

$$= 4{\cdot}4 \times 10^{-6} \text{ B.Th.U.}$$

Now the heat actually supplied to the copper is given by the specific heat $\times$ mass $\times$ rise of temperature,

i.e. $0{\cdot}09 \times 0{\cdot}318 \times 100 = 2{\cdot}86$ B.Th.U.

The heat, equivalent to the external work done, is therefore only a minute fraction of the heat which has to be supplied. The difference is the heat which goes to increase the energy of the molecules of the copper corresponding to the higher temperature. The energy of molecular motion and of configuration is called the *internal* energy of the body.

The heat supply is therefore required for two purposes: (*a*) to change the internal energy; (*b*) to do the external work which accompanies the change.

Let δQ be the total energy supplied *to* the copper,
δW, the external work done *by* the copper,
and δE, the increase in the internal energy of the copper;

then, by the conservation of energy,

$$\delta Q = \delta E + \delta W, \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

where energy, both thermal and mechanical, is measured in a common unit—say the B.Th.U.

$$\text{Since } \delta W = \frac{p\delta v}{J},$$

$$\delta Q = \delta E + \frac{p\delta v}{J}. \quad \ldots\ldots\ldots\ldots\ldots(4)$$

It is important to note just where we have made appeal to experiment in the above reasoning.

1. The copper expands when heated (by experiment), therefore external work is done. This is shown, by mechanics, to be

$$p\delta v \text{ ft-lb.}$$

2. By the first law of thermodynamics we can convert this energy into equivalent heat energy by dividing by 778, i.e.

$$p\delta v \text{ (foot-pounds) becomes } \frac{p\delta v}{J} \text{ (B.Th.U.).}$$

3. As we are now able to express heat energy and mechanical energy in a common unit—the B.Th.U.—we can assert that the *mathematical equation*

$$\delta Q = \delta E + \frac{p\delta v}{J}$$

is true, by the principle of the *Conservation of Energy*.

Although the $p\delta v$ term is minute in the above example, it is very far from being minute in the case of the gases and vapours, which we shall now discuss.

The State of a Gas or Vapour.—It is necessary to have a clear idea of what is meant in thermodynamics by the *state* of a gas or vapour. We suppose ourselves to be considering a given portion of matter which we call the "system", the "body", or the "working substance". A knowledge of this "system" comes from certain impressions which it makes on our senses either directly or through our instruments of observation. Our information of the system is "complete" when we know all its qualities or properties, such, for instance, as its position, density, state of strain, chemical composition, temperature, &c. Some of these properties are understood to be given to us as part of the data of the question; for instance, in steam engineering we usually think of a pound of steam. We also imagine it to be at the earth's surface and to be at rest as a whole, that is to say, we do not take into account any kinetic energy which the pound of steam may possess in virtue of the motion as a whole. Many other properties, however, are variable, such as the temperature, volume, electrical state, &c. It is the object of thermodynamics to find how far *changes* in the state of a body are determined by **mechanical and thermal** changes to which the body may be subjected. Before we can attack this problem it is clear we must choose a system of quantities which will define the state of the body, just as before we draw a map of a new country we must consider exactly how we are going to measure our lines to determine accurately the position of each place; and further, we must have **datum states** in thermodynamics just as we must have datum lines in ordinary surveying. In mathematical language we must have a suitable set of co-ordinates and a *frame of reference* in which to frame our picture.

We shall suppose that the body is a permanent gas and that we know how much is given, suppose 1 lb. We shall also suppose that it is unelectrified and remains so, it is non-magnetic and remains so—in fact,

we shall consider only changes in the *mechanical* and *thermal state* of the gas which accompany *mechanical* and *thermal* exchanges of energy between it and its surroundings.

The most convenient quantities to measure, a knowledge of which gives the mechanical and thermal states of a gas, are its *pressure*, *volume*, and *temperature*. If, however, we know the mass of gas, we do not need to measure each of these, for, given any two, say the volume and temperature, the third can be calculated from

$$\frac{PV}{M} = \frac{R}{m}T,$$

where M is the mass of gas in pounds,
P is the pressure in pounds per square foot,
V is the volume in cubic feet,
R is the universal gas constant (p. 135),
m the molecular weight of the gas,
and T the absolute temperature in degrees Fahrenheit ($460 + t°$ F.).

It is sufficient, therefore, to study the changes in any *two* of the three quantities P, V, and T, say V and T, which follow exchanges of heat and mechanical energy. We shall therefore take V and T as the *independent variables* the changes in which we have to calculate.

A Vapour: Wet or Superheated.—Even if the substance is a vapour, we can still define the state of the substance by specifying two quantities, its volume and temperature,* but the problem is not so simple as that of a permanent gas. We shall consider the case of steam.

1. By experiment, we know the *saturation pressure* for the vapour at any given temperature. This information is given in a steam-table.

2. We also have Callendar's equation for steam, viz.:

$$V = 0{\cdot}595\frac{T}{P} - 0{\cdot}421\left(\frac{671{\cdot}6}{T}\right)^{\frac{10}{8}} + 0{\cdot}016.$$

Fix on any temperature—say 212° F., i.e. 671·6° F. absolute (taking $T = 459{\cdot}6 + t$, instead of $T = 460 + t$), and we get

$$V = \frac{399{\cdot}5}{P} - 0{\cdot}405. \qquad \ldots\ldots\ldots\ldots\ldots\ldots \text{(A)}$$

The steam-table gives 14·69 lb. per square inch as the saturation pressure corresponding to 212° F. Plot the curve represented by equation (A) from P = 15 to P = 500, say, and draw a line parallel to the V axis

* In dealing with wet steam we must choose the two quantities carefully. For instance, if we chose temperature and pressure, we should leave the volume indeterminate if the pressure happened to be the saturation pressure for the temperature in question. In these circumstances pressure and temperature are *not* independent of each other, and so cannot form a pair of *independent* variables. To determine the volume, we should have to know the dryness fraction. Provided we choose a pair of really *independent* variables, all is well—volume and temperature is such a pair.

at $P = 14{\cdot}69$, to cut the curve. We thus get a line ABCD in fig. 21 (*a*). This line is the "isothermal" for steam at 212° F. Taking a series of temperatures, we can construct an "isothermal map", as it were, such as is sketched in fig. 21 (*b*).

Now suppose we are given the volume and temperature of the steam, say V_2 and T_2. We know its *state-point* lies on the T_2 isothermal. We

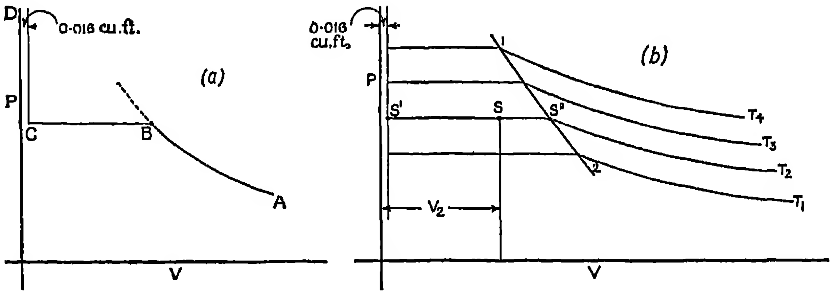

Fig. 21

know its volume V_2, and the only possible state-point is therefore at S, where the volume and temperature have the given values. The pressure is then read off—it is P_2, say, the saturation pressure at T_2—and the dryness fraction q is given by the ratio $S'S/S'S''$. We have thus found the pressure, volume, temperature, and dryness fraction of the steam by specifying the *two* quantities, volume and temperature.

To determine changes in these two quantities which accompany exchanges of heat and mechanical energy, we use an extremely ingenious method of reasoning, due to Sadi Carnot, which is based on the idea of reversible cyclic changes.

Cyclic Changes.—By a "cyclic change" is meant a change in which the working substance is taken through a series of changes of state starting from an initial state defined by P_0, V_0, T_0, q_0, and returning finally to exactly the same state, defined by P_0, V_0, T_0, q_0. The state of the substance at the beginning and end of the cycle is therefore exactly the same. Any quality or property of the working substance which depends *only* on its state must thus be unaltered in amount in a cyclic change.

A Reversible Exchange.—The agencies which we are going to use to cause the working substance to change its state are the agencies of heat and mechanical energy. For instance, we can warm a gas and make it expand; we can mechanically compress a gas and make it contract. We will consider first a reversible exchange of heat. If the working substance is throughout at temperature T, and we bring a hot body in contact with it (say a hot electric wire immersed in the gas) at temperature $T + \epsilon$, heat will flow from the body at $T + \epsilon$ into the working substance at T provided ϵ is positive. If ϵ is negative the reverse will happen. By a reversible exchange of heat we mean an exchange of heat under conditions

such that ϵ is infinitesimally small, so that a very slight change in the temperature of either body will reverse the direction in which heat flows. In other words, we suppose that the temperatures of the two bodies are equal except for the minute difference which is necessary to ensure a flow of heat from one body to the other, when the working substance is absorbing or emitting heat.

The idea of a reversible exchange of mechanical energy follows naturally. Suppose we have the gas in a cylinder with a very light frictionless piston, and suppose that the pressure outside the piston exactly balances the pressure of the gas on the inside of the piston. The gas on the inside of the piston is our working substance. If we make the pressure on the outside of the piston higher by a minute amount, the gas will be slowly compressed and work will be done *on* the gas by the external force on the piston. On the other hand, if we make the pressure on the outside of the piston very slightly less than the pressure of the gas, the gas will drive the piston out and do work against the external pressure on the piston. In the former case the energy of the working substance will be increased because it receives mechanical energy from outside sources; in the second case the energy of the working substance will be decreased because the working substance does work against external force.

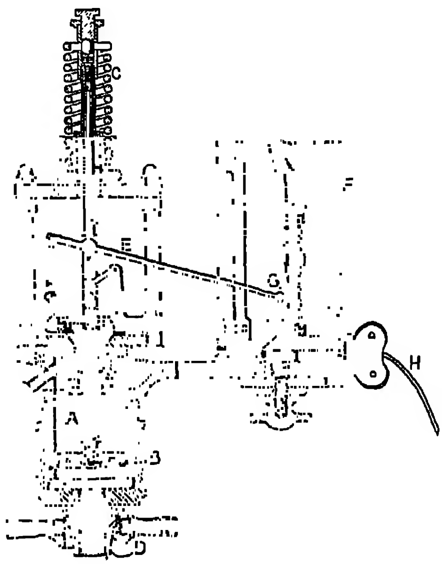

Fig. 22

A Reversible Cyclic Change.—A reversible cyclic change is a cyclic change in which all exchanges of heat and mechanical energy are reversible ones.

An Irreversible Cyclic Change.—An irreversible cyclic change is a cyclic change in which the exchanges of mechanical energy and heat between the working substance and external bodies are not reversible. Both kinds of cycles can be arranged, but, as we shall see later, the reversible cycle has important properties.

Mechanical Work done in a Reversible Cyclic Change.—The work done by a gas which is subjected to a cyclic change in a cylinder can be measured by the *indicator*, the instrument used in steam engineering to draw mechanically a p, v diagram for the steam on one side of a piston.

A is a small cylinder (fig. 22), B an accurately fitted piston of 1 sq. in., say, in area, C a spring, D a joint by means of which the instrument can

be attached to the engine cylinder, E a link mechanism by which the movement of the piston B is repeated at the pencil G, and F a drum on which a piece of glazed paper is carried. The drum F is given a to-and-fro rotational motion by means of a driving cord H, the free end of which is attached to the crosshead of the engine through a reducing mechanism. The string is kept taut by working against a spiral spring contained in the drum F. The motion of F therefore repeats the outward and inward motion of the engine piston on a small scale; while the vertical motion of the pencil G, across the paper on F, repeats the motion of the piston B on a magnified scale. The pressure in the engine cylinder is the pressure on the steam side of B. The piston B therefore responds, against the spring, to changes in the cylinder pressure, and the pencil G rises and falls as this pressure rises and falls.

The pencil therefore traces out a graph like that shown in fig. 23.

The curve is closed, since the changes are cyclic and steady.

The area enclosed by this curve is a measure of the actual work done on one side of the piston.

The appropriate scale factor f is known when the conditions under which the instrument is used are known, and the work done, in foot-pounds, is the product of the scale factor by the area enclosed by the curve, in square inches.

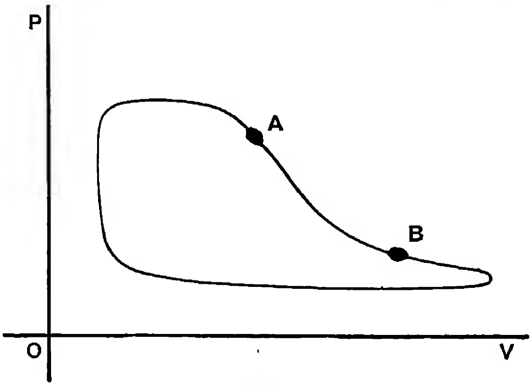

Fig. 23

Of course any consistent units can be used by giving different values to f.

Exchange of Heat and Work in a Reversible Cycle.—Suppose fig. 23 represents the indicator diagram for a certain mass m of steam (wet or dry or superheated) on one side of a single-acting engine piston.

The working substance clearly goes through a cyclic change, the initial and final states being identical as regards pressure, volume, temperature, and dryness.

Clearly work is done *by* the working substance against external forces if the diagram is traversed in the clockwise direction, and this work is proportional to the area of the indicator diagram. This amount of energy must therefore have left the working substance. By the law of the conservation of energy this energy must have been derived from somewhere, i.e. some body or bodies must have supplied it. There are only three possibilities.

(*a*) The steam has been absorbing heat from the furnace and emitting heat to the condenser or the atmosphere; if the former exceeds the latter, there will have been a net input of heat during the cycle, and this may

supply the whole of the energy taken out of the working substance mechanically during the cycle.

(*b*) The internal energy of the working substance itself may have been tapped.

(*c*) Perhaps a combination of both (*a*) and (*b*) explains where the energy comes from.

The first hypothesis is the correct one.

We can easily see that (*b*) is an untenable hypothesis. The state of the working substance at the end of the cycle is exactly the same as it was at the beginning of the cycle, and we can repeat the cycle as often as we wish. If we suppose the internal energy of the working substance has been reduced by an amount E, then in going through two cycles it would be reduced by 2E, and in going through N cycles by NE. We can make N as large a number as we wish, and consequently we have the paradox that we can obtain just as much energy as we wish from a given portion of matter without in any way changing its physical state.

Conversely, if the internal energy were to be increased by an amount E per cycle, we should have the equally paradoxical result that we could pump just as much energy as we liked into the working substance without in any way changing its state. These deductions absolutely contradict our experience of the properties of matter. We therefore conclude that as the initial and final states of the working substance are identical in going through a cyclic change, any mechanical work which may have been done cannot have been obtained by tapping the internal energy of the working substance.

We must therefore fall back on hypothesis (*a*), and by the principle of the conservation of energy and of the mechanical equivalent of heat we are led to the conclusion that the mechanical work done in the cycle is equal numerically to the mechanical equivalent of the excess of the heat absorbed by the working substance from outside sources over that emitted by the working substance to outside sources. In other words, if Q_1 is the heat absorbed by the working substance from external sources, and Q_2 the heat emitted by the working substance to external sinks, and W the work done in mechanical units, then

$$Q_1 - Q_2 = \frac{W}{J}.$$

This equation embodies the principles of the mechanical equivalent of heat, the conservation of energy, and the properties of a cyclic change of substance. It is often referred to as the first law of thermodynamics.

Mathematically, the result can be put thus:—$\Delta Q = \Delta E + \Delta W$, where ΔQ is a small quantity of heat absorbed by the working substance from external sources, ΔE, the corresponding increase in the internal energy of the working substance, and ΔW the small amount of work done by the working sub-

stance in consequence of the absorption of heat ΔQ. In going round the cyclic change then we have

$$(\int)dQ = (\int)dE + (\int)dW.$$

But $(\int)dE = 0$, since E is a function of *the state* of the substance,

hence $(\int)dQ = (\int)dW$,

i.e. $Q_1 - Q_2 = W$, in heat units throughout.

In calculating the *absorption* of heat from external sources it must be remembered that emission of heat to external bodies is looked upon as negative absorption. The principle that the working substance possesses a definite internal energy which is a function of the state of the substance gives us one of the two equations we want, to determine the two independent variables (p. 193). To determine the remaining equation we have to make use of the *Second Law of Thermodynamics*. Before dealing with this law, it will be necessary to consider two important types of expansion to which a gas or vapour may be subjected. We are led to do this by the following considerations:

1. We have found that E is a function of state only and is thus independent of the manner in which the substance passes from state A to B.
2. We desire to find *another* function possessing the same property and so we are led to consider the *simplest* ways in which the passage from state A to state B can be effected. By studying these simple cases we may get a hint of the existence of the new function we are seeking.

Isothermal Expansion.—Suppose we have unit mass of a gas or vapour enclosed in a cylinder, and suppose we allow it to expand by altering the external pressure on the piston very slowly. In these circumstances there will be plenty of time for the substance in the cylinder to take up the temperature of the room in which we do the experiment. This temperature we suppose to be constant. From the gas equation,

$$PV = \frac{R}{m}T,$$

we see that if T is kept constant, PV must be constant, i.e. a gas expands or contracts under the conditions of Boyle's Law. Such changes of state *at constant temperature* are called isothermal changes.

Adiabatic Changes.—Now suppose that instead of giving the gas plenty of time to accommodate itself to the temperature of the room, we perform the expansion very quickly. We will suppose the expansion is a reversible one, so that the pressure opposing the expansion is just slightly less than the pressure of the gas at any particular instant. The gas evidently does work against the external mechanical force which opposes the outward motion of the piston. Consequently the energy of the work-

ing substance must decrease by a certain definite amount. The expansion is supposed to take place so quickly that there is no time for any heat to flow into the gas from the cylinder walls. Consequently the working substance does not receive a supply of energy in the form of heat from external bodies. We see that the working substance has done external work in expanding and has not received any compensating supply of energy from outside sources. Falling back on our bed-rock of the conservation of energy, it is clear that its own energy must have decreased. Consequently its temperature must have dropped.

Expansion under these conditions is thus entirely different from expansion under isothermal conditions. The condition obtaining in this type of expansion is that heat energy neither enters nor leaves the working substance. Such an expansion is called an adiabatic* expansion. The curves connecting pressure and volume for adiabatic expansions are very much steeper than those connecting pressure and volume for isothermal expansion because of the cooling of the gas in the former type of expansion.

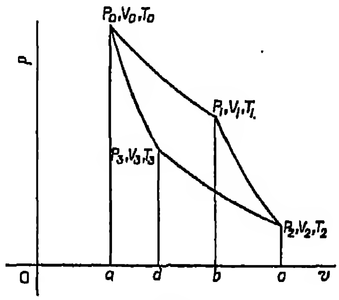

Fig. 24

Carnot's Cycle.—With these two types of expansion, we can easily plan a cycle to which we can subject *any* working substance. Suppose the working substance is initially in the state defined by P_0, V_0, T_0. We can let the substance expand isothermally from P_0, V_0, T_0 to P_1, V_1, T_1. The second step will be to let it expand adiabatically from P_1, V_1, T_1 to P_2, V_2, T_2. We can next compress the working substance from P_2, V_2, T_2 to P_3, V_3, T_3 isothermally, and we can stop isothermal compression at the right values of P_3, V_3, and T_3, so that we can bring the working substance by an adiabatic compression to its initial state P_0, V_0, T_0. This cycle is the famous one known as Carnot's cycle, and it will be seen that it consists of an isothermal expansion and compression and an adiabatic expansion and compression. The isothermal expansion and compression are distinguished by the fact that the temperature is constant during each of them, while the adiabatic expansion and compression are distinguished by their being no exchange of heat between the working substance and the outside bodies during these changes. Fig. 24 shows an ordinary indicator diagram for such a cycle, the working substance being a gas. We will consider in detail how this cycle could be brought about with a gas as working substance.

Suppose we have a gas confined in a cylinder (fig. 25),† the sides of which are impervious to heat and the end perfectly pervious to heat; we will suppose the piston to be a frictionless, massless one, and we will suppose that we have three external bodies available, a hot body at T_1, from which

* ἀ, not; διά, through; βαίνειν, to go.

† Ignore the right-hand (irreversible) cylinder in fig. 25 in this discussion.

we can extract any amount of heat we like without changing its temperature, a cold body at T_2 into which we can pump any amount of heat we wish without raising its temperature, and an insulator E_1, which we can apply to the end of the cylinder to prevent any heat passing through it. We will suppose the gas is initially at the temperature T_1 of the hot body, and that it occupies a volume V_1 at a pressure P_1. We will now suppose that we apply the hot body to the cylinder end. If the temperature of the hot body is a minute fraction of a degree higher than the temperature of the working substance, heat will pass into it *reversibly*. We will therefore suppose that a quantity of heat Q_1 passes into the gas from the hot body at T_1, and that the working substance expands isothermally. Now remove the hot body and apply the insulator, and allow the gas to expand still farther adiabatically by reducing the pressure on the outside of the piston, so that the pressure outside the piston is always just less than the pressure of the gas. This again is a reversible mechanical expansion. Now apply the cold body to the cylinder end and compress the gas isothermally by making the external mechanical force on the piston very slightly greater than the pressure of the gas. Heat will be rejected to the cold body under these isothermal conditions, since work is being done on the gas and its temperature is not allowed to change. Suppose Q_2 is the heat rejected to the cold body. We will stop this compression at the correct point, so that when we remove the cold body from the cylinder and apply again the insulator, we can bring the working substance back again to its initial state by a reversible adiabatic compression. The gas has now gone through a complete cycle. The heat taken in from the hot body was Q_1, the heat emitted to the cold body was Q_2. The internal energy of the gas is finally exactly the same as it was initially. Consequently $Q_1 - Q_2 = \frac{W}{J}$, where W is the work done during the cycle. The *efficiency* of this cycle is defined by the fraction

$$\frac{Q_1 - Q_2}{Q_1}.$$

We now have to calculate the work done in a Carnot cycle.

Carnot's Principle.—We have defined the efficiency of a heat engine as the fraction

$$\frac{Q_1 - Q_2}{Q_1}.$$

Carnot's principle is that *the test of maximum efficiency is reversibility.* Provided an engine is working on a reversible cycle between T_1 and T_2 it has the maximum possible efficiency. The reader must carefully remember that in this theorem we are talking about given temperatures T_1 and T_2, and the efficiencies of different engines between these limits of temperature. When we say that the reversible cycle is that of maximum

efficiency we mean that it has the maximum efficiency of any engine working between these temperatures.

Carnot Cycle for Wet Steam.—Fixing now upon two temperatures T_1 and T_2, we will think of *any substance* which goes through a reversible cycle between these limits of temperature, i.e. the cycle is arranged so that the substance takes in all the heat it absorbs at the *highest* temperature, and throws out all the heat it emits at the *lowest* temperature. For instance, if we were working with wet steam we should have an isothermal absorption of heat—the absorption of the latent heat of the steam at T_1. This would be followed by an adiabatic expansion to the lower temperature and to a lower pressure. Then would follow compression and condensation at the lower temperature until a certain point was reached from which a final adiabatic compression would convert the steam and water into water at the high temperature. This cycle is clearly a pure Carnot's cycle.

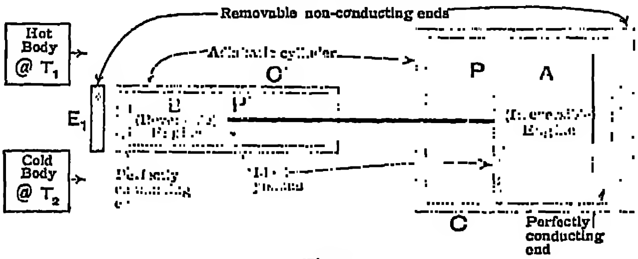

Fig. 25

We will now suppose that we have a second engine (fig. 25) which works on an irreversible cycle. We have as yet proved nothing as to the efficiency of it. We shall now show that *the efficiency of such an irreversible cycle cannot possibly exceed the efficiency of a reversible cycle working between the same limits of temperature.* We will call the reversible engine B, and the irreversible engine A. A being irreversible we can only work it directly, but our first engine B is a strictly reversible one. We can thus couple it as shown in fig. 25 to the engine A, so that when A works direct, B is reversed. We have a common pair of sources and sinks of heat at T_1 and T_2. We will now suppose the two engines connected together as shown in the figure, so that A works B backwards. Suppose A absorbs an amount of heat Q_1 from the hot body, and rejects an amount of heat Q_2 to the cold body in a cycle. B is reversed so that it will reject a quantity of heat Q'_1 to the hot body and absorb a quantity of heat Q'_2 from the cold body. The efficiency of the A engine is

$$\frac{Q_1 - Q_2}{Q_1};$$

the efficiency of the B engine is*

$$\frac{Q'_1 - Q'_2}{Q'_1}$$

* For, by definition, the efficiency is $\dfrac{(-Q'_1) - (-Q'_2)}{(-Q'_1)}$.

We will suppose that the A engine—the irreversible one—is *more* efficient than B. Then

$$\frac{Q_1 - Q_2}{Q_1}$$

is greater than

$$\frac{Q'_1 - Q'_2}{Q'_1}.$$

No *external* work can be done, because one engine simply works the other, and consequently $Q_1 - Q_2$ is equal to $Q'_1 - Q'_2$; i.e. Q_1 must be less than Q'_1, and consequently Q_2 must be less than Q'_2.

Now $Q'_1 - Q_1$ is the total heat put *into* the hot body, while $Q'_2 - Q_2$ is the net heat extracted from the cold body per cycle by *both* the engines. Both these quantities are positive, since Q_1 is less than Q'_1, and Q_2 is less than Q'_2. Therefore we have the paradox that a self-contained mechanism can be constructed which does no external work, and yet causes heat to pass continuously from a cold body to a hot body, and not from a hot body to a cold body.

Second Law of Thermodynamics.—It is felt that an average state of things of this kind is impossible, and the principle of the second law of thermodynamics is put in the following form— the form in which Clausius put it: " It is impossible for a self-acting machine unaided by any external agency to convey heat from one body to another at a higher temperature ". The assumption that the irreversible engine had a higher efficiency than the reversible engine led us, as we have shown above, to a direct contradiction of this principle, and the conclusion is therefore that our initial assumption is wrong. In fact, the line of argument shows that engine A, whether irreversible or not, cannot have a *higher* efficiency than the engine B. *The maximum efficiency we can have between given upper and lower temperatures T_1 and T_2 is therefore a function of the temperatures alone and is quite independent of the nature of the working substance.* All *reversible* cycles working between given limits of temperature have this efficiency, i.e. the *maximum* efficiency. Any working substance will give this efficiency for a given pair of temperatures, provided we use a truly reversible cycle. It makes, therefore, not the slightest difference whether we calculate the efficiency of this cycle for the simplest possible case, namely, for a perfect gas, or whether we go through the much longer calculation for a vapour as the working substance. The result will be exactly the same whether we use wet steam, ammonia, a perfect gas, or even a piece of steel, provided we subject the working substance to a reversible cycle, and that *all* the heat taken in by the working substance is absorbed at the higher temperature T_1, and all given out by it is rejected at the lower temperature T_2. It is well to realize this point quite clearly. The practical efficiency of a steam-engine is less than that of a gas-engine, and it is very common to hear this fact explained as being due to the " latent heat of water "; as if the latent heat of water were a peculiar

difficulty which the steam engineer has to overcome. As we have seen above, a theoretical engine using steam on the Carnot cycle is just as efficient as the theoretical engine using a gas on the Carnot cycle, and the Carnot cycle can be approximated to just as readily with a steam-engine as with a gas-engine. The true reason of the differences in the efficiencies of practical engines will be dealt with more fully later; suffice it to say here that it is due to the fact that the internal-combustion engine uses far higher temperatures than are possible at present with the steam-engine, and therefore the ideal efficiency of the internal-combustion engine is considerably higher than the ideal efficiency of the steam-engine.

Having seen that the Carnot efficiency is a function of temperature only, and is independent of the properties of the working substance we use, we will now calculate the efficiency for an air-engine on Carnot's cycle.

Work done during Isothermal Change of Volume.—It is important to ascertain how much work is done on or by a gas when it is compressed or when it expands isothermally.

Let the original pressure and volume respectively be P_1 and V_1, then since the temperature remains constant P_1V_1 = a constant; call this constant A.

Now suppose an extremely small decrease of volume dV_1 to take place, then the work done on the gas is $P_1dV_1 = A\dfrac{dV_1}{V_1}$ where $\dfrac{dV_1}{V_1}$ is a very small fraction.

If then the process be split up into an extremely large number of such steps, we see that the total work done is the product of A × (the sum of a large number of small fractions). No two of these small fractions have the same value, so that their summation is only possible by the calculus. Integrating we have

$$\begin{aligned} W &= A\int_{V_2}^{V_1}\frac{dV}{V} \\ &= A(\log_e V_1 - \log_e V_2) \\ &= P_1V_1 \log_e \frac{V_1}{V_2} \text{ or } P_2V_2 \log_e \frac{V_1}{V_2} \\ &= P_1V_1 \text{ (or } P_2V_2) \times 2{\cdot}30 \times \log_{10} \frac{V_1}{V_2} \qquad \text{(1)} \end{aligned}$$

If P is the pressure in (*a*) pounds per square foot, (*b*) grammes per square centimetre;

and V, the volume in (*a*) cubic feet, (*b*) cubic centimetres;

then W is the work in (*a*) foot-pounds, (*b*) gramme-centimetres.

Suppose air is compressed from 8 c. ft., at atmospheric pressure, to 2 c. ft., the temperature being constant at 32° F.

$$\begin{aligned} P_1V_1 &= 14{\cdot}7 \times 144 \times 8, \\ \text{hence } W &= 14{\cdot}7 \times 144 \times 8 \times 2{\cdot}3 \times \log_{10}\tfrac{8}{2} \\ &= 14{\cdot}7 \times 144 \times 8 \times 2{\cdot}3 \times 0{\cdot}602 \\ &= 23400 \text{ ft.-lb.} \end{aligned}$$

Using the gas equation, $PV = R'T$, we get

$$W = R'T \log_e \frac{V_1}{V_2}.$$

Adiabatic Expansion.—It is found by experiment that when a gas expands adiabatically its pressure and volume are given by

$$PV^\gamma = \text{a constant.}$$

γ has different values for different gases. It is $1{\cdot}41$ for air, so we get

$$PV^{1\cdot41} = \text{a constant.}$$

Suppose the gas changes from P_1, V_1 to P_2, V_2, then

$$P_1V_1^\gamma = P_2V_2^\gamma.$$

But we *also* have the gas equation

$$\begin{aligned} \frac{P_1V_1}{T_1} &= \frac{P_2V_2}{T_2}; \\ \text{also } P_1V_1 \cdot V_1^{\gamma-1} &= P_2V_2 \cdot V_2^{\gamma-}, \\ \text{i.e. } T_1V_1^{\gamma-1} &= T_2V_2^{\gamma-1}. \quad \ldots\ldots\ldots\ldots\ldots\ldots (2) \end{aligned}$$

Efficiency of the Gas Carnot Cycle.—Let P_1, V_1, T_1 be the initial state of the gas at the beginning of the isothermal expansion. Let V_2 be the volume at the end of the isothermal expansion; then

$$W_1 = R'T_1 \log_e \frac{V_2}{V_1}.$$

Let V_3 be the volume at the end of the adiabatic expansion; then

$$\begin{aligned} T_2V_3^{\gamma-1} &= T_1V_2^{\gamma-1}. \text{ by equation (2).} \\ \therefore \frac{T_1}{T_2} &= \left(\frac{V_3}{V_2}\right)^{\gamma-1} = \left(\frac{V_4}{V_1}\right)^{\gamma-1}, \text{ similarly.} \end{aligned}$$

Joule and Thomson showed experimentally that, for the permanent gases, the internal energy depends on the temperature only—to a high order of accuracy. We may therefore suppose that in a perfect gas, $E = f(T)$. But $\Delta Q = \Delta E + \Delta W$, hence $Q = W$ when T, i.e. E, is constant, hence

$$Q_1 = \frac{R'T_1}{J} \log_e \frac{V_2}{V_1}, \text{ and } Q_2 = \frac{R'T_2}{J} \log_e \frac{V_3}{V_4}.$$

$$\text{But } \frac{V_4}{V_1} = \frac{V_3}{V_2},$$

$$\text{since } \left(\frac{V_3}{V_2}\right)^{\gamma-1} = \left(\frac{V_4}{V_1}\right)^{\gamma-1} = \frac{T_1}{T_2}.$$

$$\therefore \frac{V_3}{V_4} = \frac{V_2}{V_1}.$$

$$\therefore Q_1 = AT_1,$$

$$Q_2 = AT_2,$$

$$\text{where } A = \frac{R'}{J}\log_e\rho, \text{ and } \rho = \frac{V_2}{V_1} = \frac{V_3}{V_4}.$$

$$\therefore \text{ the efficiency } (\eta) = (Q_1 - Q_2)/Q_1$$
$$= (T_1 - T_2)/T_1.$$

The efficiency (η) of *any* reversible cycle with *any* working substance which works between T_1 and T_2 is therefore given by

$$\eta = \frac{T_1 - T_2}{T_1}.$$

Entropy.—We have seen that the efficiency of the Carnot engine is given by

$$\frac{Q_1 - Q_2}{Q_1} = \frac{T_1 - T_2}{T_1},$$

$$\text{i.e } \frac{Q_1}{T_1} = \frac{Q_2}{T_2} = \text{K, a constant.}$$

In the Carnot engine Q_1 is the heat absorbed by the working substance, and Q_2 that emitted by the working substance. If we consider only the flow of heat in one direction as positive, that is, the absorption of heat by the working substance as positive, then we should represent Q_2 by $-Q_2$, the minus sign indicating that it is an emission of heat by the working substance. In this case then we should get $\frac{Q_1}{T_1} + \frac{Q_2}{T_2} = 0$.

This equation suggests that there is a quantity which is unaltered in going round a cycle. We know there is one quantity which is unaltered by going round a cycle, namely, the internal energy. Suppose there was another quantity which we shall call ϕ, then $\Delta_1\phi$ might equal the increase in ϕ during the absorption of heat, and $\Delta_2\phi$ might be the *increase* in ϕ during the emission of heat. We should then get, if ϕ were a quantity which was unchanged during a cycle, $\Delta_1\phi + \Delta_2\phi = 0$, and it is evident from the fact that $\frac{Q_1}{T_1} + \frac{Q_2}{T_2} = 0$, that if we identify $\Delta_1\phi$ with $\frac{Q_1}{T_1}$, and $\Delta_2\phi$ with $\frac{Q_2}{T_2}$, we should have the equation $\Delta_1\phi + \Delta_2\phi = 0$.

This second quantity which remains constant round a cycle is called the *entropy* of the substance, and for *an isothermal exchange of heat* it is defined in this way: *Increase in entropy from state A to state B equals the heat absorbed by the working substance in passing from state A to state B divided by the absolute temperature at which this heat is absorbed.* This is the third principle which we referred to on p. 198 as being required to determine the state of a substance.

Summary of the Thermodynamic Principles.—We saw on p. 193 that several quantities were required to determine the state of a working substance.

1. For every substance, there is a certain definite relation between these quantities which can be expressed as an equation between them. This equation is called the *characteristic equation* of the substance. For a perfect gas it is PV = R′T, see p. 135. For steam it is Callendar's equation, namely,

$$V - \cdot 016 = 0\cdot 595\frac{T}{P} - 0\cdot 421\left(\frac{671\cdot 6}{T}\right)^{\frac{10}{3}} \quad \text{(see p. 193)}.$$

There is also a definite relation between P and T for wet steam, and q for superheated steam (p. 184).

2. The internal energy of a working substance is a function of the state of the substance only, and therefore, during a cyclic change, remains unaltered. The consequence of this fact is that the difference between the heat absorbed and the heat rejected by the working substance multiplied by Joule's mechanical equivalent of heat is numerically equal to the mechanical work done during a cyclic change.

$$\text{i.e. } (Q_1 - Q_2)J = W,$$

where Q_1 is the heat absorbed by the working substance, Q_2 the heat emitted by the working substance, J the mechanical equivalent of heat, i.e. 778 ft.-lb. per B.Th.U., and W the mechanical work done in foot-pounds during the cycle. The equation is written mathematically,

$$\delta Q = \delta E + \delta W,$$

$$\text{or } \left(\int\right)dQ = \left(\int\right)dW,$$

$$\text{since } \left(\int\right)dE = 0.$$

3. We are led to consider a thermal quantity called **the entropy**, which is defined as follows: The increase in entropy during an isothermal change of state from that defined as A to that defined as B is equal numerically to the heat absorbed by the working substance divided by the absolute temperature at which the exchange of heat takes place. The increase of entropy from the state defined by A to the state B is $\int_A^B dQ/T$,

the integral being calculated along any reversible path connecting the state denoted by A to that denoted by B. We then have the principle—during a cyclic change the entropy at the beginning of the cycle is the same as it is at the end, so that

$$\oint\frac{dQ}{T} = 0.$$

The actual properties of the working substance enter the problem in the characteristic equations. The principle numbered 2 above is variously called the principle of energy, or the first law of thermodynamics. It is really nothing more than a statement of the principle that perpetual motion against resistance is impossible.

The third principle is based on the second law of thermodynamics, and is more difficult to understand. It is intimately related with the efficiency of heat engines, and the line of argument is very briefly this. (*a*) For given limits of temperature there is a maximum efficiency. (*b*) The test for maximum efficiency is the reversibility of the cycle. (*c*) All reversible cycles working between the same limits of temperature are equally efficient. (*d*) Any irreversible cycle cannot be more efficient than a reversible one, but it may be less efficient. (*e*) The maximum efficiency is a function of the temperature only, and is quite independent of the particular properties of the working substance. It is given by the formula $\frac{T_1 - T_2}{T_1}$. (*f*) By comparing this equation with the equation to which we are directly led by the first law of thermodynamics and the definition of efficiency, namely $\frac{Q_1 - Q_2}{Q_1}$, we are led to the equation $\frac{Q_1}{T_1} + \frac{Q_2}{T_2} = 0$; and this equation will follow if we assume the existence of a property belonging to the working substance which is a function of the state of the substance only, and which is defined so that, if ϕ be this quantity,

$$\delta\phi = \frac{\delta Q}{T},$$

where $\delta\phi$ is a small increase in ϕ and δQ a small input of heat. This quantity was called the *entropy* by Clausius.

The Practical Analogy for Entropy.—The entropy function arose in the theory of heat mathematically, and engineers often prefer to have a physical basis for the quantities they use.

The idea of the entropy may perhaps be best realized by an analogy. We know by mechanics that the mechanical work done by a substance in expanding from a volume V to a volume $V + \delta V$ is given by $\delta W = P\delta V$, where P is the pressure on the substance. We have seen in the preceding pages that heat is a form of energy, and consequently it should be possible, one would think, to represent a small heat exchange δQ by an equation similar to the equation above. We also know that

temperature is what determines the direction of flow of heat. For instance, if we have a thin plate of metal with heat flowing from one edge of it to the other, we can draw over the plate a series of isothermals, and these isothermals will be arranged so that the flow of heat is everywhere perpendicular to the isothermals, and takes place in the direction in which the temperature decreases, so that the heat flows from places of high temperature to places of low temperature. Temperature then is in some ways analogous to mechanical pressure. It is in fact a kind of thermomotive force just as pressure is a ponderomotive force, and as electrical pressure is an electromotive force. Temperature, then, in the thermal formula might take the place of pressure in the mechanical formula. We are then left to find some quantity to correspond to the increase in mechanical volume. This quantity we see from the entropy equation, p. 207, is simply the increase of entropy, and we then have an exactly analogous equation, namely, $\delta Q = T\delta\phi$, in which δQ is the increase in the heat energy, T the "thermomotive force", i.e. the temperature, and $\delta\phi$ the increase in "thermal volume". For some purposes we may therefore look upon the entropy of a working substance as if it were its thermal volume. It depends, just as its real volume does, on its state. It is then quite natural that the thermal volume is unchanged in a cyclic change, and if we were to assume the existence of and to define thermal volume in this way, we could deduce Carnot's theorem and the Second Law directly.

Summary.—The principles we shall use in the succeeding engineering chapters are:

1. The characteristic properties of the substances usually employed in engines as given in tables of properties of gases, steam, ammonia, &c.

2. The principle that the internal energy of a substance is a function of its state, and hence the external work done in a cycle is equal to the excess of the heat taken in over that given out by the working substance.

3. The principle that the entropy is a function of the state of the substance. The increase of entropy between a state denoted by A and one denoted by B is defined as

$$\int_A^B \frac{dQ}{T}.$$

For steam it is reckoned from water at 32° F. as zero (state A), and for any other state the value of the integral is tabulated in the steam tables.

CHAPTER VIII

The Efficiency of the Thermal Cycles of Internal-combustion Engines

Practical Engine Cycles.—All engine cycles are irreversible both as regards mechanical and thermal exchanges; otherwise the cycle would be carried out indefinitely slowly. In expansion of the working substance, for instance, the friction of the piston must be overcome, and this is an irreversible resistance. Also, the working substance is heated and cooled very rapidly, therefore the temperature of the external body is greater than that of the working substance when heating up, and less, when cooling, *by a considerable amount.* The exchanges of heat are therefore irreversible. Further, in the case of gas- or oil-engines, the process which goes on in the cylinder is not even a true "cycle"; for the working substance burns irreversibly in the process, and therefore its initial and final "state" cannot possibly be the same. The best we can do, theoretically, is to construct a "cycle"—reversible if possible—which resembles, as closely as may be, the actual process that goes on in the cylinder or engine.

The two most important internal-combustion engine cycles are the so-called Otto and the Diesel cycles, though they are not true cycles in the strict thermodynamic sense.

Otto Cycle.—1. A combustible mixture of gas and air is sucked into a cylinder at approximately atmospheric pressure.

2. It is compressed, approximately adiabatically, to a small volume at a high pressure and a fairly high temperature.

3. It is exploded and the pressure rises more or less abruptly according to the speed of the explosion.

4. The high-pressure mixture (no longer the same, chemically, as it was in stages 1, 2, and 3) expands approximately adiabatically, doing external work and getting cooler in the process.

5. The burnt mixture at a low pressure and a fairly low temperature is expelled from the cylinder into the atmosphere.

In this cycle we get one explosion per two revolutions of the engine, and the engine is known as a four-stroke engine, i.e. four strokes go to each cycle.* The programme is repeated in the next two revolutions of the engine, and so on indefinitely. The nearest ideal cycle to this one is as follows:

Start with a volume v_1 of any permanent gas at pressure p_1, and temperature T_1, in an ideal cylinder. The suffixes refer to the states at the corresponding points on the diagrams. Trace out the following changes on the p, v diagram (fig. 26):

1. Heat the gas at constant volume by applying hot bodies at progressively

* Many writers refer to this engine as "the four-cycle engine"—"four", presumably being an abbreviation for "four-stroke". Shop practice seems to favour the term used in the text.

increasing temperatures (reversible heating) from T_1 to T_2. Its pressure will be increased.

2. Expand the high-pressure gas adiabatically and reversibly against a resisting pressure $F = p$ (reversible expansion). The gas does work, and therefore its temperature drops to T_3. Let p_3 and v_3 be the pressure and volume at the end of the expansion.

3. Cool the gas at constant volume v_3 from T_3 to T_4, by applying cold bodies of progressively decreasing temperatures (reversible cooling). The cooling is stopped at the temperature T_4, which is such that the point 4 is on the adiabatic curve through the point 1.

4. Compress the gas adiabatically and reversibly from p_4, v_4, T_4 to p_1, v_1, T_1.

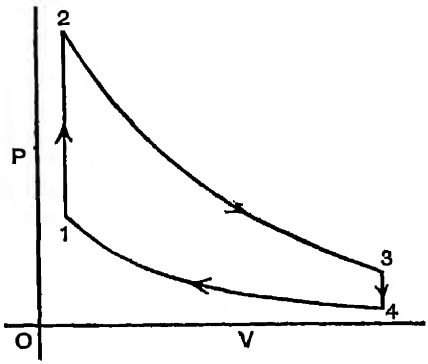

Fig. 26.

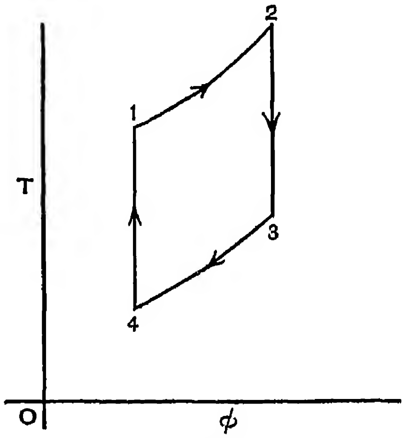

Fig. 27

This is possible since the point 4 was taken on the adiabatic through the point 1. The temperature rises during this compression, as external work is done on the gas and heat does not enter or leave it.

This cycle yields a definite amount of external work which is absorbed in a mechanical form. The p, v diagram shows a definite area which measures this available work. Now, the efficiency of the cycle is the ratio

$$\frac{\text{number of units of work done}}{\text{mechanical equivalent of heat taken in from hot bodies}}$$

and it is very important to know what this fraction (η) is, as it measures the efficiency of the cycle as a means of transforming heat energy into mechanical energy.

We can calculate this fraction by thermal methods in the following way, which is based on the use of entropy. The method is not the shortest one in this particular case, but it is simple, and is an example of a *general* method which is applicable to vapours as well as gases.

The method is based on the fact that the total change in entropy in going round any *reversible* cycle is zero.

Our imaginary cycle has been carefully made reversible so that we can apply the above rule.

Trace the various exchanges of heat and entropy as one pound of the gas goes through the cycle beginning at state 1. (See fig. 27.)

Stage.	Heat into Gas.	Increase of Entropy.
$1 \rightarrow 2$	$C_v(T_2 - T_1)$	$C_v\int_{T_1}^{T_2}\frac{dT}{T} = C_v \log_e\frac{T_2}{T_1}$
$2 \rightarrow 3$	0	0
$3 \rightarrow 4$	$-C_v(T_3 - T_4)$	$-C_v \log_e\frac{T_3}{T_4}$
$4 \rightarrow 1$	0	0

$C_v(T_2 - T_1) - C_v(T_3 - T_4)$ is the thermal value of the work done

$$\therefore \eta = \frac{C_v(T_2 - T_1 - T_3 + T_4)}{C_v(T_2 - T_1)} = 1 - \frac{T_3 - T_4}{T_2 - T_1},$$

$$\text{Now } \left(\int\right) d\phi = 0,$$

hence, adding up the right-hand column in the table, we get

$$C_v \log_e\frac{T_2}{T_1} - C_v \log_e\frac{T_3}{T_4} = 0.$$

$$\therefore \frac{T_2}{T_1} = \frac{T_3}{T_4}.$$

$$\therefore \frac{T_2}{T_1} - 1 = \frac{T_3}{T_4} - 1.$$

$$\therefore \frac{T_2 - T_1}{T_1} = \frac{T_3 - T_4}{T_4}.$$

$$\therefore \frac{T_3 - T_4}{T_2 - T_1} = \frac{T_4}{T_1} = \frac{T_3}{T_2}.$$

$$\therefore \eta = 1 - \frac{T_3 - T_4}{T_2 - T_1} = 1 - \frac{T_3}{T_2} = \frac{T_2 - T_3}{T_2} = \frac{T_1 - T_4}{T_1}.$$

Now, in fig. 26, the curved boundary lines of the cycle are adiabatic lines, i.e.

$$Tv^{\gamma-1} = \text{constant (p. 204)}.$$

$$\therefore \frac{T_4}{T_1} = \left(\frac{v_1}{v_4}\right)^{\gamma-1} = \left(\frac{1}{r}\right)^{\gamma-1},$$

$$\text{where } r = \frac{v_4}{v_1} = \text{ratio of expansion,}$$

$$\text{and } \eta = 1 - \left(\frac{1}{r}\right)^{\gamma-1}. \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

This is the formula for the efficiency of the ideal gas cycle which most nearly corresponds to the actual Otto cycle. The nature of the gas does not affect the result. It is the same for all gases which obey the gas law $pv = R'T$.

Oil-engines: Diesel Cycle.—1. A charge of air is sucked into a cylinder at approximately atmospheric pressure and temperature.

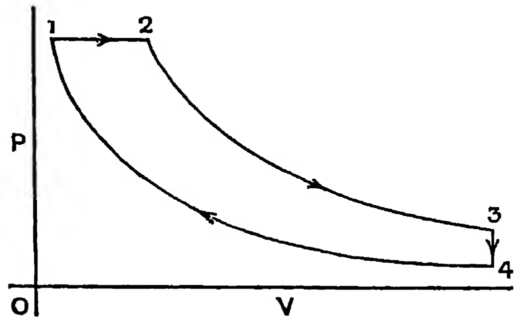

Fig. 28

2. It is highly compressed, more or less adiabatically, so that its temperature is sufficiently high for oil to burn when squirted into it.

3. By means of a pump, oil is squirted into the cylinder and burns. While burning proceeds, expansion is allowed to take place so that the pressure does not rise, i.e. the heat energy of combustion, except such as is used in the expansion referred to, goes into the gas at constant pressure.

4. When combustion is complete, the burnt mixture is allowed to expand more or less adiabatically, thereby doing work and getting cooler.

5. When the piston gets to the end of its stroke, the cylinder is connected to the atmosphere, and the contents of the cylinder, at a fairly low pressure and temperature, are expelled from the cylinder into the atmosphere during the ensuing stroke.

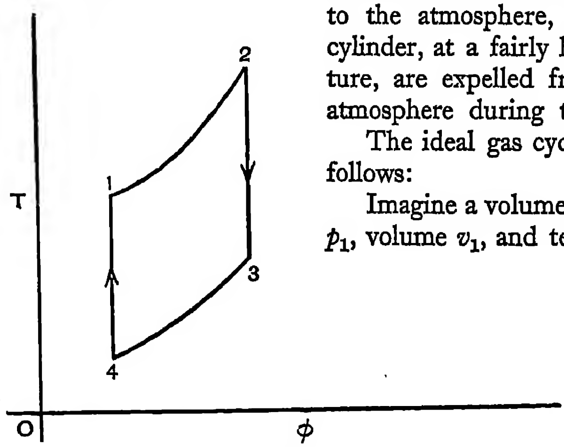

Fig. 29

The ideal gas cycle nearest to this one is as follows:

Imagine a volume of gas, initially at pressure p_1, volume v_1, and temperature T_1, in an ideal cylinder, and subject it to the following cycle (see fig. 28):—

1. Heat it at constant pressure, by means of a set of hot bodies at progressively increasing temperatures, from p_1, v_1, T_1 to p_2, v_2, T_2, keeping the pressure constant.

2. Expand it adiabatically to p_3, v_3, T_3.

3. Cool the gas at constant volume, by applying cold bodies at progressively decreasing temperatures, from p_3, v_3, T_3 to p_4, v_4, T_4 ($v_3 = v_4$).

4. Compress the gas adiabatically to its initial state, the point 4 being chosen upon the adiabatic line through 1.

The cycle is reversible, hence $(\int)d\phi = 0$.

These changes of heat and entropy per pound are as follows:

Stage.	Heat into Gas.	Increase of Entropy.
$1 \rightarrow 2$	$C_p(T_2 - T_1)$	$C_p \log_e \frac{T_2}{T_1}$
$2 \rightarrow 3$	0	0
$3 \rightarrow 4$	$-C_v(T_3 - T_4)$	$-C_v \log_e \frac{T_3}{T_4}$
$4 \rightarrow 1$	0	0

$$(\int)d\phi = 0 \text{ gives}$$

$$C_p \log_e \frac{T_2}{T_1} = C_v \log_e \frac{T_3}{T_4},$$

$$\gamma C_v \log_e \frac{T_2}{T_1} = C_v \log_e \frac{T_3}{T_4}, \text{ where } \gamma = \frac{C_p}{C_v}.$$

$$\therefore \gamma \log_e \frac{T_2}{T_1} = \log_e \frac{T_3}{T_4}.$$

$$\therefore \left(\frac{T_2}{T_1}\right)^\gamma = \frac{T_3}{T_4}.$$

The efficiency of the cycle is given by

$$\eta = \frac{C_p(T_2 - T_1) - C_v(T_3 - T_4)}{C_p(T_2 - T_1)}$$

$$= 1 - \frac{1}{\gamma}\frac{T_3 - T_4}{T_2 - T_1}$$

$$= 1 - \frac{1}{\gamma}\left\{\frac{\left(\frac{T_2}{T_1}\right)^\gamma - 1}{\frac{T_2}{T_1} - 1}\right\}\frac{T_4}{T_1}.$$

$$\text{But } \frac{p_1 v_1}{T_1} = \frac{p_2 v_2}{T_2}.$$

$$\therefore \frac{T_2}{T_1} = \frac{v_2}{v_1}, \text{ as } p_2 = p_1$$

$$= \rho, \text{ say,}$$

and T_4 and T_1 are values of T on an adiabatic line.

$$\therefore \frac{T_4}{T_1} = \left(\frac{1}{r}\right)^{\gamma - 1},$$

as proved in connection with equation (1).

$$\therefore \eta = 1 - \frac{1}{\gamma}\left\{\frac{\rho^\gamma - 1}{\rho - 1}\right\}\left(\frac{1}{r}\right)^{\gamma-1} \quad \ldots\ldots\ldots\ldots(2)$$

This formula gives the value of the efficiency required, where r is the expansion ratio v_4/v_1 and ρ is the expansion ratio at constant pressure, v_2/v_1.

In comparing this value of η with that given in (1), we have to find out whether

$$\frac{\rho^\gamma - 1}{\gamma(\rho - 1)} \text{ is } >, =, \text{ or } < 1, \text{ when } \rho > 1,$$

$$\text{i.e. whether } (\gamma - 1) \text{ is } >, =, \text{ or } < (\gamma\rho - \rho^\gamma).$$

$$\text{Put } f(\rho) = (\gamma\rho - \rho^\gamma);$$

$$\therefore f(1) = (\gamma - 1);$$

$$\text{and } \frac{d}{d\rho}f(\rho) = (\gamma - \gamma\rho^{\gamma-1}),$$

which is negative for all values of $\rho > 1$. Hence $f(\rho)$ *decreases* as ρ increases from 1.

$$\therefore (\gamma\rho - \rho^\gamma) < (\gamma - 1) \text{ when } \rho > 1, \text{ for } f(1) = (\gamma - 1).$$

$$\therefore (\gamma\rho - \gamma) < \rho^\gamma - 1 \text{ when } \rho > 1.$$

$$\therefore \left(\frac{\rho^\gamma - 1}{\gamma\rho - \gamma}\right) > 1 \text{ when } \rho > 1.$$

$$\therefore 1 - \left(\frac{1}{r}\right)^{\gamma-1} > 1 - \left(\frac{1}{r}\right)^{\gamma-1}\left\{\frac{\rho^\gamma - 1}{\gamma\rho - \gamma}\right\}.$$

This result would suggest that a higher efficiency is attainable with an Otto engine than with a Diesel. But, in practice, a very much greater ratio of compression can be used in the Diesel than in the Otto engine, for the reason that in the Diesel engine a non-explosive substance—pure air—is compressed. In consequence of this the Diesel is somewhat more, not less, efficient than the Otto engine. On the other hand, its mechanical efficiency is less, so that, on the whole, there is not a great deal of difference in the net thermal efficiencies of these engines. It must always be borne in mind that the theoretical cycles are merely the best approximations to the actual " cycles", and that these latter are limited by many practical considerations which are ignored (because they do not arise) in the theoretical treatment. The practical considerations are, of course, of first importance, as the cycles have to be actually carried out with engineering materials and appliances. Though one cycle be more efficient than another, theoretically, it by no means follows that the practical cycle corresponding to it will be more efficient than the other. It may be less efficient because of practical limitations.

CHAPTER IX

The Efficiency of the Thermal Cycles of Steam-engines and Refrigerating Machines

The T, Φ Diagram.—Suppose we have an *Indicator Diagram* giving the state of the steam on one side of the piston of a steam-engine. (Fig. 30.) This diagram is a p, v diagram. At any point A we know the pressure and volume of the steam, and hence we can calculate from the steam tables the state of the steam if we know how much of it there is in the cylinder. Or we can suppose that the cylinder has 1 lb. of steam in it and calculate on this basis. In either case, we can calculate the dryness fraction or the degree of superheat if we have an indicator diagram before us.

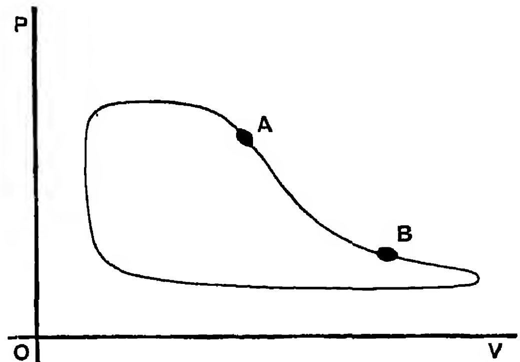

Fig. 30

Example.—The pressure and volume corresponding to the point A are 66 lb. per square inch absolute and 0·5 cu. ft. There is $\frac{1}{10}$ lb. wet steam present.

Specific volume of dry steam at 66 lb. per square inch absolute is 6·57 cu. ft.

$$\therefore \text{Volume of } \tfrac{1}{10} \text{ lb. dry steam is } 0{\cdot}657 \text{ cu. ft.}$$

$$\therefore \text{Dryness fraction} = \frac{0{\cdot}5}{0{\cdot}657} = 0{\cdot}761.^*$$

Since we can calculate the state of the steam at every point of the cycle, we can calculate *its entropy* at every point of the cycle. We also can get the temperature of the steam from the tables, since we know the pressure and volume of it. We can therefore replace the p, v diagram by a T, ϕ diagram, a diagram that is very useful in steam calculations.

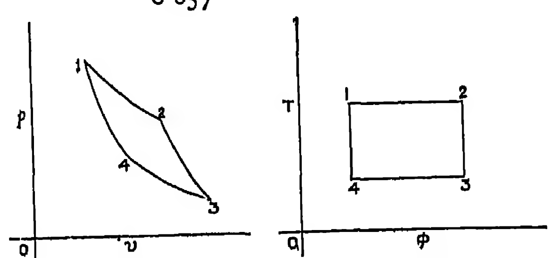

Fig. 31

* In this calculation we neglect the sp. vol. of water. If we include this, the figures would be

$$q\,6{\cdot}57 + (1 - q) \times 0{\cdot}016 = 5 \text{ (see p. 064);}$$
$$\text{i.e. } q = 0{\cdot}760.$$

Carnot's Cycle.—The p, v diagram of Carnot's cycle is given in fig. 31 *a*.

The path 1, 2 is an *isothermal* one. It is therefore a straight line 1, 2, parallel to the ϕ axis, on the T, ϕ diagram (fig. 31 *b*).

The path 2, 3 is *adiabatic*, hence the heat passing into the working substance is zero, hence

$$\Sigma \frac{dQ}{T} = 0 \text{ from 2 to 3;}$$

$$\text{i.e. } d\phi = 0 \text{ from 2 to 3;}$$

$$\text{i.e. } \phi \text{ is constant between 2 and 3.}$$

Therefore the path 2, 3 in the T, ϕ diagram is a straight line parallel to the T axis.

The T, ϕ diagram is thus simply a rectangle 1, 2, 3, 4.

$$\text{Since} \qquad \delta Q = T\delta\phi,$$

$$\left(\int\right)dQ = \left(\int\right)Td\phi; \qquad \text{(A)}$$

and the area of the T, ϕ diagram is the heat equivalent of the work done per cycle, for

$$dQ = dE + \frac{pdv}{J},$$

$$\text{and } \left(\int\right)dQ = \left(\int\right)dE + \frac{1}{J}\left(\int\right)pdv;$$

$$\text{i.e. } \left(\int\right)dQ = \frac{1}{J}\left(\int\right)pdv = \left(\int\right)Td\phi, \text{ by (A) above,}$$

$$\text{since } \left(\int\right)dE = 0.$$

The area of the rectangle 1, 2, 3, 4 is therefore the measure of the external work done per cycle.

The area of this rectangle is clearly:

$$(T_1 - T_2)(\phi_2 - \phi_1).$$

But $\phi_2 - \phi_1 = \frac{Q_1}{T_1}$ where Q_1 is the heat added to the working substance during isothermal expansion.

$$\therefore \text{ the work done is } (T_1 - T_2) \times \frac{Q_1}{T_1},$$

$$\text{i.e. } Q_1 \times \frac{(T_1 - T_2)}{T_1},$$

which is the formula we have already obtained (p. 205).

The Steam-engine Cycle.—The reader will find in the section on the steam-engine a description of the actual working of the steam-engine

in its modern form. The standard steam-cycle is called the Rankine Cycle.

Rankine's Cycle.—The steam when it leaves the boiler may be wet, dry, or, if the boiler is fitted with a superheater, superheated, i.e. at a temperature higher than the saturation temperature corresponding to the boiler pressure. These three possibilities lead to three different sets of equations, but for simplicity we shall take the single case when the steam is initially dry when it leaves the boiler. The principles upon which the calculations are based are exactly the same whether there is initial wetness or initial superheat or neither. Suppose fig. 32 is a p, v diagram representing the state of a given mass, 1 lb., say, at different points of the ideal cycle.

The point A represents the state of the substance in the boiler before its latent heat is imparted to it. We shall denote this state by p_1, ω, T_1.

The point B represents the state after evaporation (p_1, v_1, T_1).

The point C represents the state after reversible adiabatic expansion in the engine (p_2, v_2, T_2).

The point D represents the state after condensation to water in the condenser (p_2, ω, T_2).

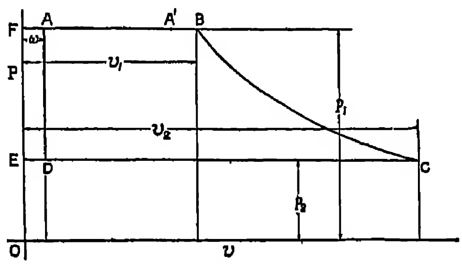

Fig. 32

We can suppose that the mass, 1 lb., goes through this cyclic change of state in a cylinder, i.e. in a "one-organ" engine.

The actual engine has several organs, namely, boiler, condenser, &c.; indeed, the idea of using several organs and keeping one organ for one purpose was Watt's great contribution to the practical development of the steam-engine. In the *ideal* theory it makes no difference in principle whether we suppose the engine to have one or several organs, but the calculation is easier to follow if we suppose the cyclic change to take place in a cylinder.

The steps in the cycle are as follows:

1. The water (at state D) is heated up from T_2 to T_1, the pressure on the piston being increased so as to correspond to the saturation pressure of the steam at each temperature.
2. Reversible evaporation under constant pressure p_1 by a hot body at temperature T_1.
3. Reversible adiabatic expansion from pressure p_1 to pressure p_2, with consequent cooling from T_1 to T_2, and some condensation.
4. Reversible complete condensation under constant pressure p_2 by a cold body at temperature T_2.

The figure ABCD (fig. 32) is the indicator diagram for this cycle, and represents the work done per pound per cycle.

The work done by the element δm is then $\delta m \times$ area ABCD $\times f$, where f is a scale factor.

The Entropy Diagram for the Rankine Cycle.—Suppose we start from the point A (fig. 32) and go round the cycle studying the exchanges of heat and entropy per pound, and the temperature.

The changes shown in the following table are easily traced.

Stage.	Heat received by Working Substance.	Increase in Entropy.
$A \rightarrow B$	L_1	L_1/T_1
$B \rightarrow C$	o	o
$C \rightarrow D$	$-q_2L_2$	$-q_2L_2/T_2$
$D \rightarrow A$	$h_1 - h_2$	$\log_e(T_1/T_2)$

We can now draw the entropy diagram (fig. 33).

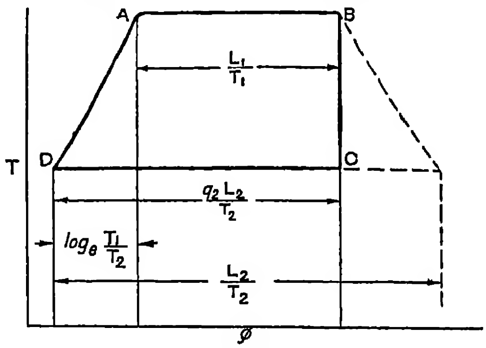

Fig. 33

The area ABCD measures the work done by the engine per cycle (p. 196), and the heat put into the steam is clearly $(L_1 + h_1 - h_2)$. We can apply the rule about the total change in entropy round a reversible cycle being zero, and hence state that

$$\frac{L_1}{T_1} - \frac{q_2L_2}{T_2} + \log_e\frac{T_1}{T_2} = 0,$$

$$\text{i.e. } \frac{q_2L_2}{T_2} = \frac{L_1}{T_1} + \log_e\frac{T_1}{T_2}.$$

This equation enables us to find the dryness fraction of the steam at C, i.e.

$$q_2 = \frac{T_2}{L_2}\left[\frac{L_1}{T_1} + \log_e\frac{T_1}{T_2}\right],$$

and q_2L_2, the heat absorbed by the condenser, is

$$T_2\left[\frac{L_1}{T_1} + \log_e\frac{T_1}{T_2}\right].$$

The difference in the heat supplied and heat absorbed per cycle per pound is the work done per cycle per pound, by the conservation of energy, hence

$$= 1 - \frac{T_2}{(L_1 + h_1 - h_2)} \times \left[\frac{L_1}{T_1} + \log_e\frac{T_1}{T_2}\right].$$

This formula gives the efficiency of the ideal Rankine cycle, using dry steam at the beginning of the expansion. It is not of much practical interest, as it is clumsy and difficult to remember, and the specific heat of water is assumed to be constant in obtaining it, which is not quite true. It is much better to study carefully the method by which the formula is obtained and to use it in the following form: Put ϕ_{w_1}, ϕ_{s_1} and ϕ_{w_2}, ϕ_{s_2} for the entropy of water and steam at T_1 and T_2 respectively. Then, by the principle of entropy,

$$q_2(\phi_{s_2} - \phi_{w_2}) = \phi_{s_1} - \phi_{w_2},$$

an equation which is at once evident from fig. 33.

$$\therefore q_2 = (\phi_{s_1} - \phi_{w_2})/(\phi_{s_2} - \phi_{w_2}),$$

and

$$\eta = \frac{(L_1 + h_1 - h_2) - L_2(\phi_{s_1} - \phi_{w_2})/(\phi_{s_2} - \phi_{w_2})}{(L_1 + h_1 - h_2)}.$$

All the quantities appearing in this formula are given in a steam table, in which allowance is made for the varying specific heat of water.

Example:—

Calculate the efficiency of Rankine's cycle for saturated steam between 200 lb./in.^{2} absolute and 2 in. Hg. condenser pressure.

At 200 lb./in.^{2} abs., $L_1 + h_1$	1205·4	by steam table.
At 2 in. Hg., h_2	67·8	
$\therefore L_1 + h_1 - h_2$	1137·6	
At 200 lb./in.^{2} abs., ϕ_{s_1}	1·5538	
At 2 in. Hg., ϕ_{w_2}	0·1292	
$\therefore (\phi_{s_1} - \phi_{w_2})$	1·4246	
At 2 in. Hg., ϕ_{s_2}	1·9767	
At 2 in. Hg., ϕ_{w_2}	0·1292	
$(\phi_{s_2} - \phi_{w_2})$	1·8475	
L_2	1033·8	by steam table.

$$\therefore \eta = 1 - \frac{1033·8}{1137·6} \times \frac{1·4246}{1·8475} = 0·30, \text{ i.e. } 30 \text{ per cent.}$$

But tables are available which give the numerator of the efficiency fraction directly, both for saturated and superheated steam (*Heat Drop Tables*, by H. Moss, D.Sc., &c., Arnold). Looking up the numerator, i.e. the so-called "heat-drop" or "work-done", we find, for 200 lb./in.^{2} saturated steam and 28 in. vacuum,

$$Q_1 - Q_2 = 338·75, \text{ by heat-drop table,}$$

$$\text{hence } \eta = 338·75/1137·6 = 0·30, \text{ 30 per cent.}$$

I, ϕ Diagrams.—It is customary to use a diagram of *total heat*, plotted against entropy, in solving practical problems of steam-engine efficiency and refrigerator performance.

On p. 182, it was mentioned that the total heat I is defined so that

$$I = E + pv,$$

where E is the internal energy, p the pressure, and v the volume, all quantities being in consistent thermal units.

The properties of I which should be noticed are:

1. During a constant-pressure process, the increase in I equals the heat added to the working substance, i.e.

$$\begin{aligned} I_2 - I_1 &= Q_2 - Q_1, \\ \text{for } \delta I &= \delta(E + pv) \\ &= \delta E + p\delta v,\ p \text{ being constant} \\ &= \delta Q \text{ (p. 191).} \\ \therefore I_2 - I_1 &= Q_2 - Q_1. \end{aligned}$$

2. During an adiabatic process, the increase in I measures the work done *on* the working substance in taking it into the cylinder at the lower pressure, compressing it adiabatically, and discharging it at the higher pressure. For

$$\begin{aligned} \delta I &= \delta E + p\delta v + v\delta p \\ &= \delta Q + v\delta p; \\ \text{and, since } \delta Q &= 0, \text{ the process being adiabatic} \\ \delta I &= v\delta p; \\ \therefore I_2 - I_1 &= \int v dp, \end{aligned}$$

the integral being taken from the state denoted by suffix (1) to that denoted by suffix (2), i.e. $I_2 - I_1$ is the area shown in fig. 34

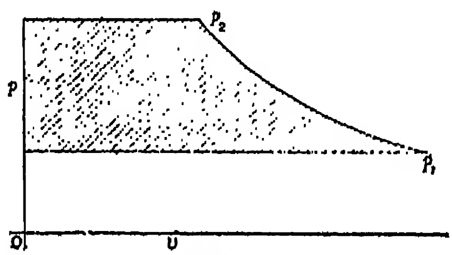

Fig. 34

3. When the working substance passes through an orifice or valve, and neither exchanges heat nor mechanical work with the "outside", I remains constant for

$$\begin{aligned} E_2 &= E_1 + p_1v_1 - p_2v_2. \\ \therefore I_2 &= I_1. \end{aligned}$$

Steam Charts.—An I, ϕ diagram for steam has been prepared by Dr. Mollier. Copies of one prepared by Professor W. E. Dalby can be obtained from Messrs. Edward Arnold. This diagram is in "pound calorie" heat units instead of B.Th.U. The shape of the diagram is seen from fig. 35.

We shall trace out the Rankine cycle for superheated steam.

Let ABCD (fig. 32) be the p, v diagram of a Rankine cycle. We start with 1 lb. of water at p_1, and at the boiling-point corresponding to p_1. The

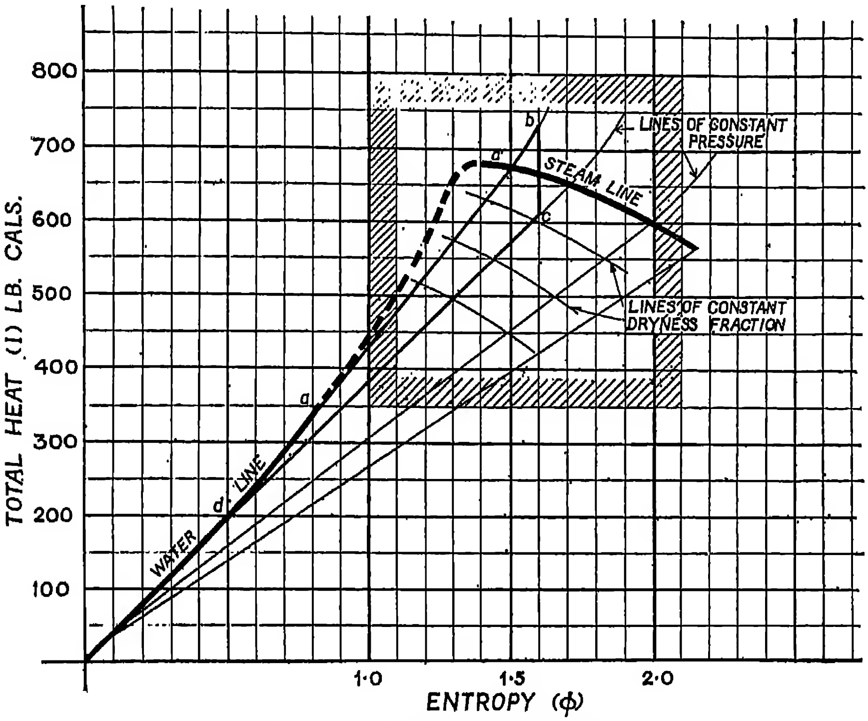

Fig. 35.—Diagram showing Rankine Cycle on Mollier Diagram for Steam

state is represented by A in fig. 32, and a in fig. 35. Evaporation at p_1 takes place, and the state becomes A′ in fig. 32 and a' in fig. 35. The path aa' is a straight line, for since p_1 is constant,

$$\begin{aligned} \delta \mathrm{I} &= \delta \mathrm{Q} \\ &= \mathrm{T}\delta\phi. \end{aligned}$$

Therefore $\dfrac{d\mathrm{I}}{d\phi} = \mathrm{T}$, a constant for evaporation at a constant pressure. The steam is then *superheated* to B (b). The adiabatic expansion (at constant entropy) brings us to C (c). The dryness fraction at c is given by the "dryness" lines which cross the fan-like evaporation lines. Condensation takes us from C to D (c to d). At d, the substance is all water at the temperature corresponding to p_2. We then apply increasing pressure again

and heat up the water from D to A (d to a). This heating up is at constant volume.

The work done is clearly measured by ABCD, and this is given by

$$(\text{FBCE} - \text{FADE}).$$

If we neglect the small area FADE, we get, since $I_2 - I_1 = \int v dp$,

$$(I_b - I_c) = \textit{work done in heat units.}$$

In ordinary steam-engine calculations, this formula is correct enough. The accurate result is:

$$\text{Work done} = \{(I_b - I_c) - \text{area FADE}.$$

The area FADE is very easily calculated directly, as it is $[(p_2 - p_1) \times 0{\cdot}016]$ in mechanical units, if the p's are in pounds per square foot;

$$\text{i.e. } (p_2 - p_1) \times \frac{0{\cdot}016 \times 144}{778} = 0{\cdot}00296\,(p_2 - p_1) \text{ B.Th.U.,}$$

when p_2 and p_1 are in pounds per square inch. The important term $(I_b - I_c)$ only remains to be obtained from the chart, and we therefore need

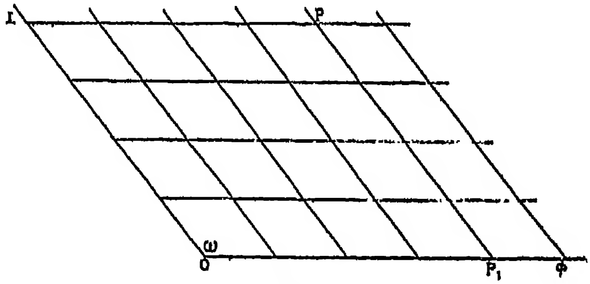

Fig. 36

only so much of the whole chart as covers the region in which the points (b) and (c) are likely to lie in practical cases, i.e. the region shaded in fig. 35. For this reason the chart is drawn only for the useful region, thus Professor W. E. Dalby's chart is drawn from $\phi = 1{\cdot}1$ to $\phi = 2{\cdot}1$, and from I = 350 lb. calories to I = 800 lb. calories.

I, ϕ Charts for Refrigerating Engineers.—The Institution of Mechanical Engineers has prepared charts for carbon dioxide, sulphur dioxide, and ammonia. Unfortunately the diagrams would be inconveniently spread out if drawn in rectangular co-ordinates, and so the diagram is slewed round by using oblique co-ordinates, i.e. we use "parallelogrammed" paper instead of "squared" paper; thus, in fig. 36,

the co-ordinates of P are OP_1 and P_1p, the angle ω being constant. This unusual system of co-ordinates makes the diagrams look more complicated than they really are.

Fig. 37 gives a general idea of the CO_2 diagram. Here, as with steam, adiabatic expansion or compression is an isentropic change of state, and hence is shown by a line *parallel to OI*, e.g. *ab* and *eg*. *abcdega* is an actual refrigerator-cycle.

I, ϕ Diagram for CO_2.—At *g* the CO_2 is all liquid at p_1. It is

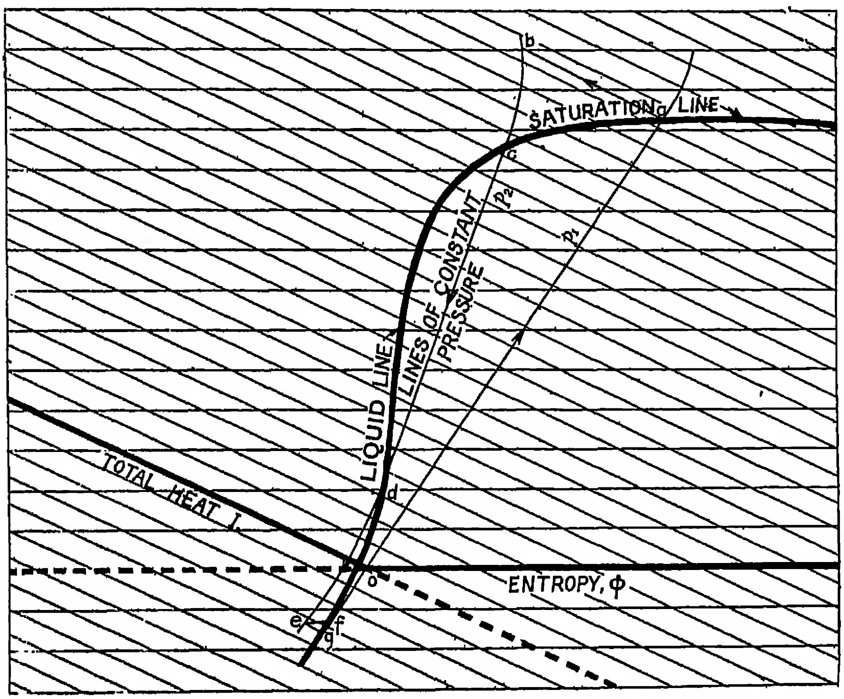

Fig. 37.—Diagram showing Refrigerator-cycle on Mollier Diagram for Carbon Dioxide

evaporated *by the cold body* and passes to state *a*. *ga* is a straight line (see p. 221).

The gaseous CO_2 is then *compressed* adiabatically, from *a* to *b*, and becomes superheated gas at p_2. It is then cooled by the *hot body* * at p_2 and passes into saturated vapour at *c*. From *c* to *d* it is condensed by the *hot body*, and at *d* is all liquid CO_2 at p_2. It is then further cooled, as a liquid, from *d* to *e*. At *e* it passes into an expansion cylinder and expands adiabatically to the lower pressure p_1. This expansion is represented by *eg*.

The cooling effect is clearly $(I_a - I_g)$.

The work done *on* the CO_2 is $[(I_b - I_a) - (I_e - I_g)] = I_\omega$.

* i.e. *hot* relative to the other outside body, but of course cold relative to the working substance. The "hot body" and the "cold body" are technical terms for the hotter and the cooler of the external sources and sink of heat.

The *coefficient of performance* is the heat removed from the cold body per unit of work done, i.e.

$$\text{Coefficient of performance} = \left[\frac{I_a - I_g}{I_\omega}\right].$$

If an expansion-valve is used instead of an expansion-cylinder, I is constant (p. 220) during this process; hence *ef* is the line followed, and

$$\text{Coefficient of performance} = \left[\frac{I_a - I_f}{I_b - I_a}\right].$$

Numerical Example.—Suppose the working substance is CO_2, the temperature of evaporation $-10°$ C., the temperature of condensation 25° C., and that 15° C. is the temperature to which the substance is cooled before passing the expansion valve.

Fig. 37 is the diagram for the ideal machine, assuming the vapour is dry at the beginning of compression. *a* is then the end of the evaporation line for $-10°$ C. The pressure of saturated vapour at 25° C. is 930 lb. per square inch,* hence *b* lies on the 930-lb. pressure line and *ab* is parallel to OI. On referring to the chart, the temperature at *b* is 63° C., hence the vapour is superheated $63 - 25 = 38°$ C., and hence can give up heat to the "hot body", i.e. the condenser at 25° C., a body "hot" relative to the evaporator at $-10°$ C.

The length of the line *ab* gives the work done in compression; it is found on the large chart for CO_2 to be equivalent to

$$I_b - I_a = 8{\cdot}72 \text{ thermal units (lb. ° C.).}$$

From *b* to *c* the gas loses its superheat along the 930 lb. per square inch line. From *c* to *d* it is being condensed at 25° C. From *d* to *e* it is being cooled as a liquid to 15° C.; *e* is therefore the intersection of the 15° C. isothermal line and the 930 lb. per square inch constant-pressure line. Draw *ef* parallel to Oϕ, to meet *ag* in *f*. The point *f* is thus fixed and we can read off the value $(I_a - I_f)$. It is

$$(I_a - I_f) = 47{\cdot}92 \text{ thermal units (by } CO_2 \text{ chart).}$$

Therefore the coefficient of performance is 47·92/8·72, i.e. 5·5.

For many similar calculations, see the "Report of the Refrigeration Research Committee", Institution of Mechanical Engineers,† October, 1914. The example given above is based on one of the calculations given in this report, which contains a Mollier diagram for carbon dioxide, for ammonia, and for sulphur dioxide, as well as a set of tables of the thermal properties of these substances. By the courtesy of the Institution of Mechanical Engineers we are able to reproduce these diagrams and tables.

* This and similar numbers are taken from the table, p. 231, and the chart for CO_2, in the pocket at the end of this volume.

† Copies can be obtained from the Institution of Mechanical Engineers, Storey's Gate, St. James's Park, London, S.W.

APPENDIX I

STEAM TABLES*

PROPERTIES OF SATURATED STEAM IN FOOT-POUND FAHRENHEIT UNITS

Abs. p. lb./in.²	Vacuum Inches.	Temperature F.		Heat in Mean British Thermal Units per Pound.			Volume per Pound.		Entropy per Pound.	
		t.	T.	h.	L.	H_s.	V_w.	V_s.	ϕ_w.	ϕ_s.
0·5	28·98	79·5	539·1	47·4	1044·7	1092·1	0·0161	640·5	0·0920	2·0299
0·55	28·88	82·4	542·0	50·3	1043·2	1093·5	0·0161	585·0	0·0974	2·0221
0·6	28·78	85·2	544·8	53·0	1041·7	1094·7	0·0161	539·1	0·1024	2·0148
0·65	28·67	87·7	547·3	55·5	1040·4	1095·9	0·0161	499·8	0·1069	2·0078
0·7	28·57	90·0	549·6	57·8	1039·2	1097·0	0·0161	466·2	0·1086	2·0018
0·75	28·47	92·2	551·8	60·0	1038·0	1098·0	0·0161	436·8	0·1151	1·9962
0·8	28·37	94·3	553·9	62·2	1036·8	1099·0	0·0161	411·1	0·1191	1·9906
0·85	28·26	96·3	555·9	64·1	1035·8	1099·9	0·0161	388·2	0·1225	1·9858
0·9	28·16	98·2	557·8	66·0	1034·8	1100·8	0·0161	367·9	0·1260	1·9810
0·95	28·06	100·0	559·6	67·8	1033·8	1101·6	0·0161	349·6	0·1292	1·9767
1·0	27·96	101·7	561·3	69·5	1032·9	1102·4	0·0161	333·1	0·1322	1·9724
1·05	27·85	103·3	562·9	71·1	1032·1	1103·2	0·0161	318·1	0·1350	1·9685
1·1	27·75	104·9	564·5	72·7	1031·2	1103·9	0·0161	304·5	0·1378	1·9646
1·15	27·65	106·4	566·0	74·2	1030·4	1104·6	0·0161	292·0	0·1405	1·9610
1·2	27·55	107·9	567·5	75·7	1029·6	1105·3	0·0161	280·6	0·1431	1·9575
1·25	27·45	109·4	569·0	77·2	1028·7	1105·9	0·0162	270·0	0·1459	1·9542
1·3	27·34	110·7	570·3	78·5	1028·0	1106·5	0·0162	260·2	0·1482	1·9509
1·35	27·24	112·0	571·6	79·8	1027·3	1107·1	0·0162	251·1	0·1504	1·9479
1·4	27·14	113·2	572·8	81·1	1026·6	1107·7	0·0162	242·7	0·1526	1·9449
1·45	27·04	114·5	574·1	82·3	1026·0	1108·3	0·0162	234·8	0·1546	1·9420
1·5	26·94	115·7	575·3	83·5	1025·3	1108·8	0·0162	227·4	0·1568	1·9392
1·6	26·73	118·0	577·6	85·8	1024·1	1109·9	0·0162	214·0	0·1608	1·9339
1·7	26·53	120·2	579·8	88·0	1022·9	1110·9	0·0162	202·2	0·1645	1·9290
1·8	26·32	122·2	581·8	90·0	1021·8	1111·8	0·0162	191·6	0·1681	1·9244
1·9	26·12	124·2	583·8	92·1	1020·6	1112·7	0·0162	182·1	0·1716	1·9200
2·0	25·91	126·1	585·7	93·9	1019·7	1113·6	0·0162	173·5	0·1748	1·9159
2·2	25·50	129·6	589·2	97·4	1017·8	1115·2	0·0162	158·7	0·1808	1·9081
2·4	25·10	132·9	592·5	100·7	1015·9	1116·6	0·0162	146·4	0·1863	1·9010
2·6	24·69	135·9	595·5	103·7	1014·3	1118·0	0·0163	135·6	0·1918	1·8947
2·8	24·28	138·8	598·4	106·6	1012·8	1119·4	0·0163	126·5	0·1962	1·8888

* The *Abridged Callendar Steam Tables, Fahrenheit Units,* published by Edward Arnold & Co., London, have been used in compiling this table, by permission of the publishers.

The columns V_w and ϕ_w were specially calculated for this table from the formulæ:

$$V_w = 0{\cdot}01602 + 0{\cdot}000023\ G,$$
$$\text{and } \phi_w = (h + G)/T,$$

where G is the "water-potential", a quantity given in Professor Callendar's table.

Abs. p. lb./in.²	Vacuum Inches.	Temperature F.		Heat in Mean British Thermal Units per Pound.			Volume per Pound.		Entropy per Pound.	
		t.	T.	h.	L.	H_s.	V_w.	V_s.	ϕ_w.	ϕ_s.
3·0	23·87	141·5	601·1	109·3	1011·3	1120·6	0·0163	118·6	0·2005	1·8833
3·2	23·46	144·0	603·6	111·8	1009·9	1121·7	0·0163	111·6	0·2047	1·8780
3·4	23·06	146·4	606·0	114·2	1008·6	1122·8	0·0163	105·4	0·2088	1·8731
3·6	22·65	148·7	608·3	116·5	1007·3	1123·8	0·0163	99·93	0·2125	1·8685
3·8	22·24	150·9	610·5	118·7	1006·1	1124·8	0·0163	95·00	0·2162	1·8641
4·0	21·83	153·0	612·6	120·8	1004·9	1125·7	0·0163	90·54	0·2198	1·8600
4·2	21·42	155·0	614·6	122·8	1003·8	1126·6	0·0163	86·50	0·2229	1·8561
4·4	21·01	156·9	616·5	124·7	1002·7	1127·4	0·0164	82·80	0·2259	1·8524
4·6	20·60	158·8	618·4	126·6	1001·6	1128·2	0·0164	79·42	0·2289	1·8489
4·8	20·20	160·6	620·2	128·3	1000·7	1129·0	0·0164	76·31	0·2318	1·8455
5·0	19·79	162·3	621·9	130·0	999·8	1129·8	0·0164	73·44	0·2346	1·8422
5·2	19·38	163·9	623·5	131·7	998·8	1130·5	0·0164	70·80	0·2372	1·8391
5·4	18·97	165·5	625·1	133·4	997·8	1131·2	0·0164	68·34	0·2400	1·8361
5·6	18·56	167·1	626·7	134·9	997·0	1131·9	0·0164	66·05	0·2424	1·8331
5·8	18·16	168·6	628·2	136·4	996·1	1132·5	0·0164	63·91	0·2447	1·8303
6·0	17·75	170·1	629·7	137·9	995·3	1133·2	0·0164	61·91	0·2473	1·8277
6·5	16·73	173·6	633·2	141·4	993·3	1134·7	0·0164	57·44	0·2527	1·8214
7·0	15·70	176·9	636·5	144·7	991·4	1136·1	0·0165	53·59	0·2579	1·8158
7·5	14·68	180·0	639·6	147·8	989·6	1137·4	0·0165	50·24	0·2626	1·8101
8·0	13·66	182·9	642·5	150·8	987·8	1138·6	0·0165	47·30	0·2670	1·8049
8·5	12·64	185·7	645·3	153·6	986·2	1139·8	0·0165	44·69	0·2717	1·8001
9·0	11·62	188·3	647·9	156·3	984·6	1140·9	0·0165	42·36	0·2760	1·7956
9·5	10·60	190·8	650·4	158·7	983·3	1142·0	0·0166	40·27	0·2797	1·7914
10·0	9·58	193·2	652·8	161·1	981·9	1143·0	0·0166	38·39	0·2834	1·7874
10·5	8·56	195·6	655·2	163·4	980·6	1144·0	0·0166	36·68	0·2868	1·7836
11·0	7·54	197·8	657·4	165·7	979·2	1144·9	0·0166	35·11	0·2903	1·7799
11·5	6·51	199·9	659·5	167·9	977·9	1145·8	0·0166	33·68	0·2938	1·7765
12·0	5·49	202·0	661·6	169·9	976·7	1146·6	0·0166	32·37	0·2968	1·7731
12·5	4·47	204·0	663·6	171·9	975·5	1147·4	0·0166	31·15	0·2997	1·7699
13·0	3·45	205·9	665·5	173·8	974·4	1148·2	0·0167	30·03	0·3026	1·7669
13·5	2·43	207·8	667·4	175·8	973·2	1149·0	0·0167	28·99	0·3055	1·7640
14·0	1·41	209·6	669·2	177·6	972·2	1149·8	0·0167	28·02	0·3083	1·7611
14·5	0·39	211·3	670·9	179·4	971·1	1150·5	0·0167	27·11	0·3112	1·7584
14·689	Gauge lb./in.²	212·0	671·6	180·0	970·7	1150·7	0·0167	26·79	0·3118	1·7573
15	0·31	213·0	672·6	181·0	970·2	1151·2	0·0167	26·27	0·3133	1·7557
16	1·31	216·3	675·9	184·4	968·1	1152·5	0·0167	24·73	0·3184	1·7506
17	2·31	219·5	679·1	187·5	966·2	1153·7	0·0168	23·37	0·3228	1·7458
18	3·31	222·4	682·0	190·5	964·4	1154·9	0·0168	22·16	0·3274	1·7414
19	4·31	225·2	684·8	193·4	962·6	1156·0	0·0168	21·06	0·3316	1·7373
20	5·31	228·0	687·6	196·1	961·0	1157·1	0·0168	20·08	0·3355	1·7333
21	6·31	230·6	690·2	198·7	959·4	1158·1	0·0168	19·18	0·3394	1·7294
22	7·31	233·1	692·7	201·3	957·8	1159·1	0·0169	18·37	0·3430	1·7258
23	8·31	235·5	695·1	203·7	956·3	1160·0	0·0169	17·62	0·3465	1·7223
24	9·31	237·8	697·4	206·1	954·8	1160·9	0·0169	16·93	0·3501	1·7189
25	10·31	240·1	699·6	208·4	953·3	1161·7	0·0169	16·29	0·3534	1·7157
26	11·31	242·2	701·8	210·5	952·0	1162·5	0·0169	15·71	0·3562	1·7126
27	12·31	244·4	704·0	212·7	950·7	1163·4	0·0169	15·16	0·3593	1·7097
28	13·31	246·4	706·0	214·8	949·3	1164·1	0·0170	14·66	0·3623	1·7069
29	14·31	248·4	708·0	216·8	948·1	1164·9	0·0170	14·18	0·3652	1·7042

Pressure lb./in.²		Temperature F.		Heat in Mean British Thermal Units per Pound.			Volume per Pound.		Entropy per Pound.	
Abs.	Gauge.	t.	T.	h.	L.	H_s.	V_w.	V_s.	ϕ_w.	ϕ_s.
30	15·31	250·3	709·9	218·8	946·8	1165·6	0·0170	13·74	0·3679	1·7016
31	16·31	252·2	711·8	220·7	945·6	1166·3	0·0170	13·38	0·3706	1·6991
32	17·31	254·0	713·6	222·5	944·4	1166·9	0·0170	12·94	0·3732	1·6966
33	18·31	255·8	715·4	224·4	943·2	1167·6	0·0170	12·57	0·3758	1·6943
34	19·31	257·6	717·2	226·1	942·1	1168·2	0·0171	12·22	0·3782	1·6919
35	20·31	259·2	718·8	227·9	940·9	1168·8	0·0171	11·90	0·3808	1·6897
36	21·31	260·9	720·5	229·5	939·9	1169·4	0·0171	11·59	0·3829	1·6874
37	22·31	262·5	722·1	231·2	938·8	1170·0	0·0171	11·29	0·3852	1·6852
38	23·31	264·1	723·7	232·8	937·8	1170·6	0·0171	11·02	0·3876	1·6831
39	24·31	265·7	725·3	234·4	936·8	1171·2	0·0171	10·75	0·3895	1·6811
40	25·31	267·2	726·8	235·9	935·8	1171·7	0·0171	10·50	0·3917	1·6792
42	27·31	270·2	729·8	239·0	933·8	1172·8	0·0172	10·03	0·3960	1·6754
44	29·31	273·0	732·6	241·9	931·8	1173·7	0·0172	9·608	0·3999	1·6719
46	31·31	275·7	735·3	244·7	930·0	1174·7	0·0172	9·212	0·4037	1·6685
48	33·31	278·4	738·0	247·4	928·2	1175·6	0·0172	8·858	0·4074	1·6651
50	35·31	280·9	740·5	250·0	926·5	1176·5	0·0173	8·520	0·4109	1·6620
52	37·31	283·4	743·0	252·6	924·8	1177·4	0·0173	8·213	0·4144	1·6589
54	39·31	285·8	745·4	255·0	923·2	1178·2	0·0173	7·928	0·4177	1·6561
56	41·31	288·2	747·8	257·4	921·5	1178·9	0·0173	7·663	0·4208	1·6533
58	43·31	290·4	750·0	259·7	920·0	1179·7	0·0174	7·415	0·4238	1·6506
60	45·31	292·6	752·2	262·0	918·4	1180·4	0·0174	7·184	0·4269	1·6479
62	47·31	294·8	754·4	264·2	916·9	1181·1	0·0174	6·966	0·4297	1·6453
64	49·31	296·8	756·4	266·4	915·4	1181·8	0·0174	6·761	0·4327	1·6429
66	51·31	298·9	758·5	268·4	914·0	1182·4	0·0174	6·571	0·4353	1·6405
68	53·31	300·9	760·5	270·5	912·6	1183·1	0·0175	6·388	0·4381	1·6382
70	55·31	302·8	762·4	272·5	911·2	1183·7	0·0175	6·218	0·4407	1·6359
72	57·31	304·7	764·3	274·4	909·9	1184·3	0·0175	6·056	0·4432	1·6337
74	59·31	306·6	766·2	276·4	908·5	1184·9	0·0175	5·902	0·4460	1·6315
76	61·31	308·4	768·0	278·3	907·2	1185·5	0·0175	5·757	0·4483	1·6294
78	63·31	310·2	769·8	280·1	905·9	1186·0	0·0176	5·618	0·4506	1·6275
80	65·31	311·9	771·5	281·9	904·7	1186·6	0·0176	5·847	0·4529	1·6256
82	67·31	313·6	773·2	283·7	903·4	1187·1	0·0176	5·362	0·4552	1·6237
84	69·31	315·3	774·9	285·4	902·2	1187·6	0·0176	5·241	0·4575	1·6218
86	71·31	317·0	776·6	287·1	901·0	1188·1	0·0176	5·127	0·4595	1·6200
88	73·31	318·6	778·2	288·8	899·8	1188·6	0·0176	5·018	0·4618	1·6183
90	75·31	320·2	779·8	290·4	898·7	1189·1	0·0177	4·913	0·4638	1·6165
92	77·31	321·7	781·3	292·0	897·5	1189·5	0·0177	4·813	0·4658	1·6148
94	79·31	323·3	782·9	293·7	896·3	1190·0	0·0177	4·717	0·4681	1·6131
96	81·31	324·8	784·4	295·2	895·2	1190·4	0·0177	4·624	0·4700	1·6115
98	83·31	326·3	785·9	296·8	894·1	1190·9	0·0177	4·535	0·4721	1·6098
100	85·31	327·7	787·3	298·3	893·0	1191·3	0·0177	4·451	0·4739	1·6082
105	90·31	331·2	790·8	302·0	890·3	1192·3	0·0178	4·251	0·4787	1·6044
110	95·31	334·7	794·3	305·6	887·7	1193·3	0·0178	4·070	0·4831	1·6007
115	100·31	338·0	797·6	309·0	885·2	1194·2	0·0179	3·903	0·4878	1·5972
120	105·31	341·1	800·7	312·3	882·8	1195·1	0·0179	3·751	0·4914	1·5938
125	110·31	344·2	803·8	315·6	880·3	1195·9	0·0179	3·609	0·4957	1·5906
130	115·31	347·2	806·8	318·7	878·0	1196·7	0·0180	3·470	0·4995	1·5875
135	120·31	350·1	809·7	321·8	875·7	1197·5	0·0180	3·358	0·5032	1·5846
140	125·31	353·0	812·6	324·8	873·4	1198·2	0·0180	3·245	0·5067	1·5818
145	130·31	355·7	815·3	327·7	871·3	1199·0	0·0181	3·140	0·5102	1·5791

Pressure lb./in.²		Temperature F.		Heat in Mean British Thermal Units per Pound.			Volume per Pound.		Entropy per Pound.	
Abs.	Gauge.	t.	T.	h.	L.	H_s.	V_w.	V_s.	ϕ_w.	ϕ_s.
150	135·31	358·4	818·0	330·5	869·2	1199·7	0·0181	3·041	0·5137	1·5765
155	140·31	361·0	820·6	333·2	867·2	1200·4	0·0181	2·949	0·5171	1·5740
160	145·31	363·5	823·1	335·9	865·1	1201·0	0·0181	2·862	0·5202	1·5715
165	150·31	366·0	825·6	338·5	863·1	1201·6	0·0182	2·781	0·5235	1·5691
170	155·31	368·4	828·0	341·1	861·1	1202·2	0·0182	2·703	0·5267	1·5668
175	160·31	370·7	830·3	343·6	859·2	1202·8	0·0182	2·631	0·5297	1·5643
180	165·31	373·0	832·6	346·1	857·3	1203·4	0·0183	2·562	0·5327	1·5620
185	170·31	375·3	834·9	348·5	855·4	1203·9	0·0183	2·496	0·5355	1·5598
190	175·31	377·5	837·1	350·9	853·5	1204·4	0·0183	2·435	0·5384	1·5577
195	180·31	379·7	839·3	353·2	851·7	1204·9	0·0183	2·376	0·5411	1·5557
200	185·31	381·8	841·4	355·5	849·9	1205·4	0·0184	2·320	0·5438	1·5538
205	190·31	383·9	843·5	357·7	848·2	1205·9	0·0184	2·266	0·5464	1·5520
210	195·31	386·0	845·6	359·9	846·5	1206·4	0·0184	2·216	0·5489	1·5502
215	200·31	388·0	847·6	362·0	844·8	1206·8	0·0184	2·167	0·5514	1·5483
220	205·31	390·0	849·6	364·2	843·1	1207·3	0·0185	2·120	0·5539	1·5465
225	210·31	391·9	851·5	366·3	841·4	1207·7	0·0185	2·076	0·5564	1·5447
230	215·31	393·8	853·4	368·4	839·7	1208·1	0·0185	2·034	0·5590	1·5429
235	220·31	395·7	855·3	370·4	838·2	1208·6	0·0185	1·993	0·5610	1·5412
240	225·31	397·6	857·2	372·4	836·6	1209·0	0·0186	1·954	0·5633	1·5395
245	230·31	399·4	859·0	374·4	835·0	1209·4	0·0186	1·916	0·5645	1·5379
250	235·31	401·2	860·8	376·3	833·4	1209·7	0·0186	1·880	0·5679	1·5362
260	245·31	404·7	864·3	380·1	830·4	1210·5	0·0187	1·811	0·5723	1·5332
270	255·31	408·1	867·7	383·8	827·4	1211·2	0·0187	1·748	0·5766	1·5303
280	265·31	411·4	871·0	387·4	824·4	1211·8	0·0187	1·689	0·5808	1·5274
290	275·31	414·6	874·2	391·0	821·5	1212·5	0·0188	1·634	0·5848	1·5246
300	285·31	417·8	877·4	394·4	818·7	1213·1	0·0188	1·583	0·5884	1·5219
310	295·31	420·8	880·4	397·8	815·9	1213·7	0·0189	1·534	0·5924	1·5192
320	305·31	423·8	883·4	401·1	813·2	1214·3	0·0189	1·489	0·5962	1·5167
330	315·31	426·7	886·3	404·3	810·6	1214·9	0·0189	1·446	0·5999	1·5142
340	325·31	429·6	889·2	407·4	808·0	1215·4	0·0190	1·406	0·6033	1·5109
350	335·31	432·4	892·0	410·6	805·3	1215·9	0·0190	1·368	0·6067	1·5096
360	345·31	435·1	894·7	413·6	802·8	1216·4	0·0191	1·333	0·6101	1·5074
370	355·31	437·8	897·4	416·5	800·4	1216·9	0·0191	1·298	0·6132	1·5053
380	365·31	440·4	900·0	419·5	797·9	1217·4	0·0191	1·266	0·6166	1·5032
390	375·31	443·0	902·6	422·3	795·5	1217·8	0·0192	1·235	0·6195	1·5012
400	385·31	445·5	905·1	425·2	793·1	1218·3	0·0192	1·206	0·6229	1·4991
410	395·31	448·0	907·6	428·0	790·7	1218·7	0·0192	1·178	0·6258	1·4971
420	405·31	450·5	910·1	430·7	788·4	1219·1	0·0193	1·152	0·6290	1·4952
430	415·31	452·9	912·5	433·4	786·2	1219·6	0·0193	1·127	0·6317	1·4933
440	425·31	455·2	914·8	436·0	784·0	1220·0	0·0193	1·102	0·6346	1·4915
450	435·31	457·5	917·1	438·7	781·7	1220·4	0·0194	1·079	0·6374	1·4897
460	445·31	459·8	919·4	441·2	779·5	1220·7	0·0194	1·057	0·6401	1·4880
470	455·31	462·1	921·7	443·8	777·3	1221·1	0·0194	1·036	0·6428	1·4863
480	465·31	464·3	923·9	446·3	775·1	1221·4	0·0195	1·016	0·6455	1·4846
490	475·31	466·5	926·1	448·8	773·0	1221·8	0·0195	0·996	0·6483	1·4830
500	485·31	468·6	928·2	451·2	771·0	1222·2	0·0195	0·977	0·6509	1·4814
510	495·31	470·7	930·3	453·6	768·9	1222·5	0·0196	0·959	0·6534	1·4798
520	505·31	472·8	932·4	456·0	766·8	1222·8	0·0196	0·942	0·6558	1·4783
530	515·31	474·9	934·5	458·3	764·8	1223·1	0·0196	0·925	0·6585	1·4768
540	525·31	476·9	936·5	460·7	762·7	1223·4	0·0197	0·909	0·6609	1·4753
550	535·31	478·9	938·5	463·0	760·8	1223·8	0·0197	0·893	0·6633	1·4738

NOTE ON THE USE OF THE STEAM TABLE

The table gives the properties of saturated steam. If superheated steam is in question, we can easily make the necessary rough adjustments by means of the following formulæ:

1. *Specific Volume.*—Calculate this quantity directly from Callendar's equation, p. 183, where T is the actual steam temperature in degrees F. absolute. It is not the "superheat", which is the difference between the steam temperature and the saturation temperature.

Example.—Find the specific volume of steam at 200 lb./in.^{2} and with 200° F. of superheat.

$$T = 841{\cdot}4 + 200 = 1041{\cdot}4.$$

$$\therefore V = 0{\cdot}595 \times \frac{1041{\cdot}4}{200} - 0{\cdot}421 \times \left(\frac{671{\cdot}6}{1041{\cdot}4}\right)^{\frac{10}{3}} + 0{\cdot}016.$$

If only a rough result is needed, the second and third term may be ignored, and the volume calculated by Boyle's Law, $PV = 0{\cdot}595T$.

2. *The Superheat.*—The additional heat imparted to the steam depends on the specific heat of steam, which is variable. For rough purposes we may take it at 0·5, so that:

$$\text{Additional heat} = 0{\cdot}5 \times \text{"degree of superheat"}.$$

Example.—Steam at 200 lb./in.^{2} with 200° F. superheat.

Total heat of saturated steam ...	1205·4
Additional heat	100·0
Total heat	1305·4

3. *Entropy of Superheat.*—The increase of entropy, due to superheat, is given roughly by

$$0{\cdot}5 \log_e \frac{T}{T_s}, \text{ i.e. by } 1{\cdot}15 \log_{10} \frac{T}{T_s},$$

where T is the steam temperature and T_s the saturation temperature, both in degrees F. absolute.

Example.—Steam at 200 lb./in.^{2} with 200° F. superheat.

Entropy of saturated steam, ...	1·5538
Increase due to superheat, $1{\cdot}15 (\log_{10} 1041{\cdot}4 - \log_{10} 841{\cdot}4)$	·1065
Total entropy	1·6603

If higher accuracy is required in superheat calculations, the special superheat tables must be used, such as those of Callendar or Marks and Davis (see p. 183).

The following calculations may be useful:

4. *To find the Correct Size of a Steam Pipe.*—The size depends on the quantity and state of the steam to be transmitted.

Suppose the pipe is to feed the high-pressure cylinder of a 1000-h.p. marine engine with steam at 150 lb./in.^{2} absolute (i.e. 135 lb./in.^{2} gauge), the engine taking 15 lb. of steam per horse-power hour. Then:

Steam to be passed, per hour, 15×1000	15,000 lb.
Steam to be passed, per second, $\frac{15{,}000}{60 \times 60}$ ...	$\frac{25}{6}$ lb.
Specific volume of steam at 150 lb./in.^{2} ...	3·04 c. ft.
Volume of steam passing, per second, $\frac{25}{6} \times 3{\cdot}04$	12·65 c. ft.
* Speed of steam allowed, feet per second ...	100.
Area of pipe required, $\frac{12{\cdot}65}{100}$	0·1265 sq. ft.
Internal diameter of pipe in feet	0·4 ft.
Internal diameter of pipe in inches, say ...	5 inches.

* Usual steam speeds are:

High-pressure saturated steam	100 ft./sec.
High-pressure superheated steam	150 ft./sec.
Exhaust steam:	
Reciprocating engines	100 150 ft./sec.
Turbines	400 ft./sec.

5. A 2000-h.p. marine engine used 15·0 lb. of steam per indicated horse-power hour, the steam being supplied as saturated steam at 160 lb. per square inch gauge, and the vacuum being 23·9 in. Hg. Calculate the Rankine efficiency, the actual thermal efficiency, and the efficiency ratio for the engine.

Proceeding exactly as shown on p. 219, we find that the heat input per pound $(L_1 + h_1 - h_2)$ is 1093·5, and the Rankine efficiency is 24·9 per cent.

The actual efficiency is obtained thus:

1 i.h.p. hour is equivalent to a cylinder-output of 2545 B.Th.U. of energy, hence the actual efficiency is:

$$\frac{2545}{15 \times 1093{\cdot}5} = 15{\cdot}5 \text{ per cent,}$$

since the steam used per indicated horse-power hour is 15·0 lb., and so the heat input per indicated horse-power hour is $15{\cdot}0 \times 1093{\cdot}5$.

The efficiency-ratio is $\frac{15{\cdot}5}{24{\cdot}9}$, i.e. 62·25 per cent.

APPENDIX II

Charts and Tables showing Properties of Substances used in Refrigeration

(By permission of the Institution of Mechanical Engineers)

Carbonic Acid

Pressure.	Temperature.		Volume of Saturated Vapour.
Pounds per sq. in.	° C.	° F.	Cubic ft. per lb.
200	− 32·0	− 25·5	0·463
225	− 28·3	− 18·8	0·409
250	− 24·8	− 12·7	0·365
275	− 21·7	− 7·0	0·330
300	− 18·7	− 1·7	0·301
325	− 16·0	+ 3·2	0·276
350	− 13·4	+ 7·9	0·255
375	− 10·9	+ 12·3	0·236
400	− 8·6	+ 16·5	0·219
450	− 4·3	+ 24·3	0·191
500	− 0·3	+ 31·4	0·169
550	+ 3·4	+ 38·1	0·150
600	+ 6·8	+ 44·2	0·134
650	+ 10·0	+ 50·0	0·120
700	+ 13·0	+ 55·4	0·108
800	+ 18·6	+ 65·5	0·0878
900	+ 23·6	+ 74·6	0·0720
1000	+ 28·3	+ 82·9	0·0565
1050	+ 30·5	+ 86·8	0·0460
1070	+ 31·3	+ 88·4	0·0352

Ammonia

Pressure.	Temperature.		Volume of Saturated Vapour.
Pounds per sq. in.	° C.	° F.	Cubic ft. per lb.
6	− 49·2	− 56·6	41·6
8	− 44·2	− 47·6	31·6
10	− 40·2	− 40·4	25·75
12	− 36·8	− 34·3	21·75
14	− 33·8	− 28·9	18·79
16	− 31·2	− 24·1	16·56
18	− 28·8	− 19·8	14·82
20	− 26·6	− 15·9	13·45
25	− 21·8	− 7·2	10·88
30	− 17·7	+ 0·1	9·17
35	− 14·2	+ 6·5	7·93
40	− 11·0	12·2	6·99
45	− 8·1	17·4	6·25
50	− 5·5	22·1	5·66
60	− 0·83	30·5	4·77
70	+ 3·3	37·9	4·12
80	+ 6·9	44·5	3·63
90	10·3	50·5	3·25
100	13·3	56·0	2·936
120	18·8	65·8	2·466
140	23·6	74·5	2·124
160	27·9	82·3	1·868
180	31·9	89·4	1·666
200	35·5	95·9	1·504
225	39·6	103·2	1·340
250	43·4	110·1	1·208
275	46·9	116·4	1·098
300	50·2	122·4	1·007
350	56·2	133·2	0·863
400	61·6	142·9	0·752
450	66·6	151·9	0·665
500	71·1	160·0	0·597
550	75·3	167·6	0·539
600	79·3	174·7	0·491
650	83·0	181·4	0·449
700	86·5	187·7	0·414

Sulphurous Acid

Pressure.	Temperature.		Volume of Saturated Vapour.
Pounds per sq. in.	° C.	° F.	Cubic ft. per lb.
5	− 32·0	− 25·6	14·1
6	− 28·6	− 19·5	11·9
7	− 25·8	− 14·4	10·3
8	− 23·2	− 9·7	9·04
9	− 20·8	− 5·4	8·10
10	− 18·6	− 1·5	7·35
12	− 14·7	+ 5·6	6·17
14	− 11·2	+ 11·8	5·34
16	− 8·2	+ 17·3	4·70
18	− 5·4	+ 22·3	4·19
20	− 2·8	+ 26·9	3·80
25	+ 2·7	+ 36·9	3·08
30	+ 7·4	45·4	2·59
35	11·6	52·8	2·24
40	15·3	59·5	1·97
45	18·6	65·5	1·76
50	21·6	71·0	1·59
60	27·1	80·8	1·33
70	31·9	89·3	1·15
80	36·1	97·0	1·01
90	40·0	103·9	0·90
100	43·5	110·3	0·82

STEAM BOILERS

BY

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Steam Boilers

Introductory

Although the various types of modern steam boilers differ considerably in detail, they may be classified broadly into three main types, namely, vertical, horizontal smoke-tube, and water-tube boilers. Vertical boilers may be either of the smoke-tube or water-tube type. The choice of any particular type depends on a large number of conditions, among which may be mentioned the steam pressure and rate of evaporation required, the nature of the load, and the available floor space and head room. Other conditions which require careful consideration include the nature of the fuel and water available, while under certain circumstances the matter of transportation may be a determining factor.

With other conditions fixed and a given grate area, a certain rate of burning fuel will give a maximum of economy. The ideal boiler should transfer the maximum possible amount of the heat so generated to the water in the boiler, consistent with economy. At first sight it appears, therefore, that the heating surface of the boiler, that is, the area which is in contact with the water on one side and the hot gases on the other, should be made as large as possible. It must be remembered, however, that after a certain point the cost of extending the heating surface will more than compensate for the heat lost in the waste gases. There is therefore a certain ratio of heating surface area to grate area which will give a maximum of economy, this ratio obviously depending, among other things, on the particular type of boiler under consideration.

CHAPTER I

Vertical Boilers

The vertical type of boiler, generally limited to small sizes, has the advantages of being self-contained and of cheapness. Vertical boilers are more or less portable, are easily accommodated, and, in general, raise steam quickly. Owing to absence of brickwork and their portability, they are especially

useful for temporary installations. Although not of high economic efficiency, yet, as weight is small for power supplied, for many purposes a good vertical boiler made by a reputable firm is cheap and convenient. While its construction permits of it being made for any reasonable pressure, the water capacity being somewhat limited, some attention is required to ensure a steady steam supply, and owing to limited steam space there is a liability to priming. The simple vertical boiler, consisting of a fire-box surrounded by water contained in a cylindrical shell, although raising steam quickly yet also burns fuel at a high rate, and various modifications have been made in order to increase its efficiency.

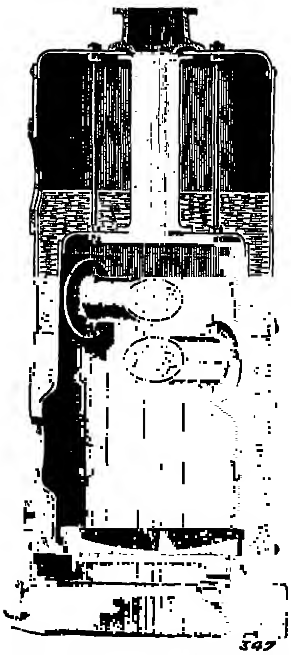


Fig. 1.—Vertical Cross-tube Boiler

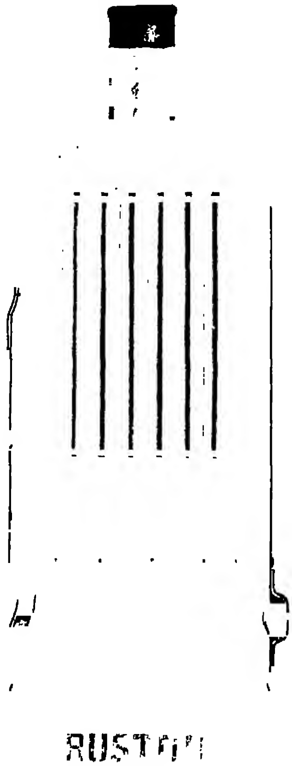


Fig. 2.—Vertical Boiler, Type L.T.

One of the simplest forms of vertical boiler is shown in fig. 1. The two cross tubes increase the heating surface area, improve circulation and also baffle the furnace gases to some extent. This type is simple and easy to clean, but uses rather a great amount of fuel. Greater efficiency is obtained by the type of boiler shown in fig. 2, where the furnace gases must pass through a series of vertical smoke tubes before reaching the chimney. Good water should be used, otherwise the tubes will become coated with a non-conducting scale, which will soon lead to burning out of the tubes. One objection to this design is that the upper portions of the tubes are not surrounded by water, and are therefore liable to overheating and corrosion.

A water-tube vertical boiler is shown in fig. 3. This type possesses the advantages of comparatively large heating surfaces, improved circulation, and good access for examination and cleaning. Two sides of the fire-box are flattened to form tube plates, and a nest of inclined water tubes is placed across the fire-box. A manhole at each end of the tubes facilitates the removal of scale from the inside of the tubes, or removal of any tube if necessary. This type is a quick steamer and comparatively economical in the use of fuel, about 7 lb. of water being evaporated per pound of coal. It will stand

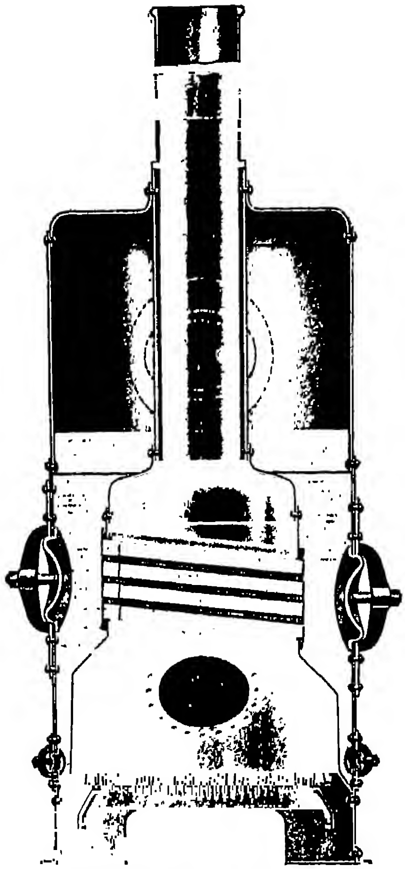


Fig. 3.—Vertical Water-tube Boiler

forcing, the rate of evaporation being increased if desired by a steam jet in the uptake. A squat pattern of this type used for launches is shown in fig. 4. Fig. 5 shows a design incorporating the use of "Field" tubes. These tubes consist of an outer and inner tube, the outer tube being closed at the bottom and open at the top, whilst the inner tube is open both top and bottom. They hang from the crown of the fire-box and are full of water. The idea of the tube is that the outer tube being in contact with the hot furnace gases, the water will circulate down the inside of the inner tube, and up between the inner and outer tubes. If, however, the firing is too great or the water is dirty this circulation is not continuous, and the tubes are liable to be burnt out.

A good design of vertical boiler is shown in fig. 6, which gives a sectional

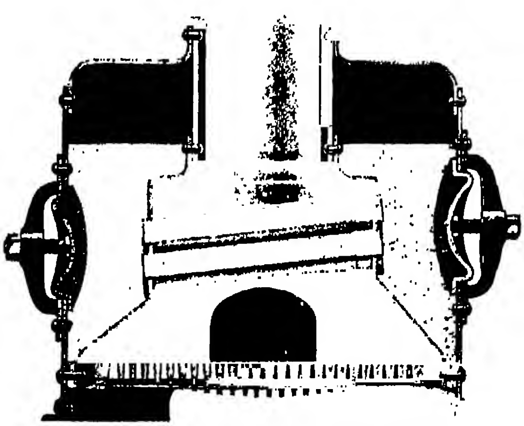


Fig. 4.—Vertical Water-tube Boiler, Launch Pattern

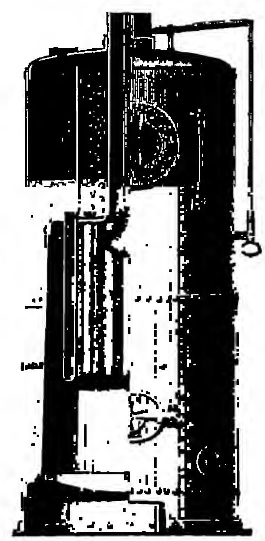


Fig. 5.—Vertical Boiler with Field Tubes

view of the Cochran boiler. The furnace dome is seamless, being pressed from one plate to finished shape. There are therefore no riveted seams

exposed to the flames. From the furnace the products of combustion pass into a combustion chamber at the back of the boiler, then through a number of horizontal and parallel smoke tubes to the smoke-box at the front. The tubes, which vary in number from about 50 to 280, are about $2\frac{1}{2}$ in. external diameter, and being completely surrounded by water, any tendency to leak, due to undue variation in temperature, is eliminated. The large number of tubes gives great heating surface, and tends to increase efficiency. The tubes can easily be seen and cleaned when the smoke-box doors are opened. Both smoke-box and back plate are easily removed, in fact, the whole design is based on complete accessibility combined with good efficiency. These boilers will burn from 15 lb. of coal per square foot of grate per hour in the smallest, to 24 lb. in the largest boiler, and the water evaporated per pound of coal varies from 5 to about $6\frac{1}{2}$ lb. A group of these boilers is shown in fig. 7. The standard sizes of these

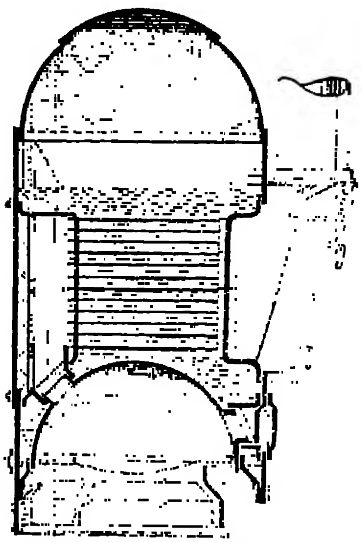


Fig. 6.—Cochran Vertical Boiler (section)

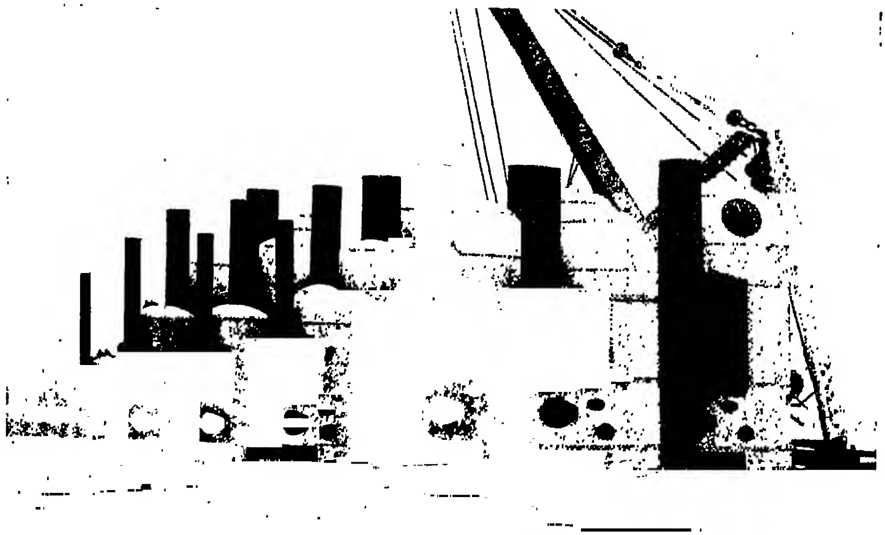


Fig. 7.—Group of Cochran Vertical Boilers

boilers vary from 3 ft. to $8\frac{1}{2}$ ft. in diameter, with an evaporation according to size of from some 300 to 6000 lb. per hour.

The Paxman "Essex" Boiler is designed on somewhat similar lines.

The furnace gases are conveyed into a combustion chamber at the side, where they are divided and pass through curved tubes to a similar chamber on the other side provided with a door and uptake, constituting a smoke-box. The tubes are somewhat semicircular in shape, and the curve of the tubes being of a greater radius than the boiler, their removal is easy. The shape of the tubes permits access to the top of the fire-box for inspection and cleaning purposes.

A vertical boiler designed for a particular purpose is shown in section in fig. 8, which illustrates the Merryweather water-tube safety boiler. The design is such that a very large heating surface with a corresponding grate area in proportion to the over-all dimensions is obtained. The boiler consists of a fire-box and shell, and outside fire-box plate between which and the fire-box is a water space. The shell is bolted above to the uptake and below to the outside firebox plate by means of two angle rings, the joints on the uptake being of asbestos, and on the lower ring of rubber held in a groove. Removal of this shell to expose the tubes for examination and cleaning is a comparatively simple process. The water tubes are of two kinds, the majority being straight and slightly inclined from the horizontal, while the others are bent at one end and are arranged vertically, these latter increasing the water circulation considerably. The tubes are generally of solid-drawn copper, although they can be made of steel if required. The shape of the bottom part of the fire-box permits of sufficient grate area being obtained. Induced draught can be obtained by supplying exhaust steam through a baffle into the uptake, or by an auxiliary blast pipe fitted into the uptake through which live steam is supplied. Forced draught by means of a fan can also be arranged for. The feature of this boiler is its capacity of standing forcing, and its ability to raise steam quickly from cold water. The makers claim that under special conditions of firing, steam can be obtained in ten minutes without risk of injury to the boiler. With induced draught an evaporation of 15 to 20 lb. of water per square foot of heating surface per hour can be obtained. This feature, together with its great portability, makes it especially suitable for portable installations where quick steaming is required, such as in fire engines, salvage pumps, steam vehicles, &c. An exterior

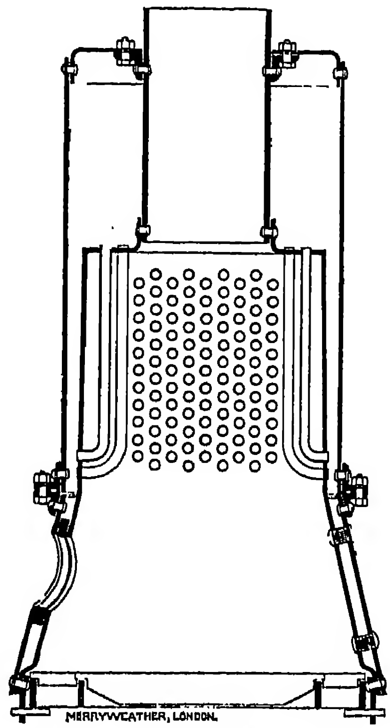


Fig. 8.—Standard Merryweather Vertical Water-tube Boiler (section)

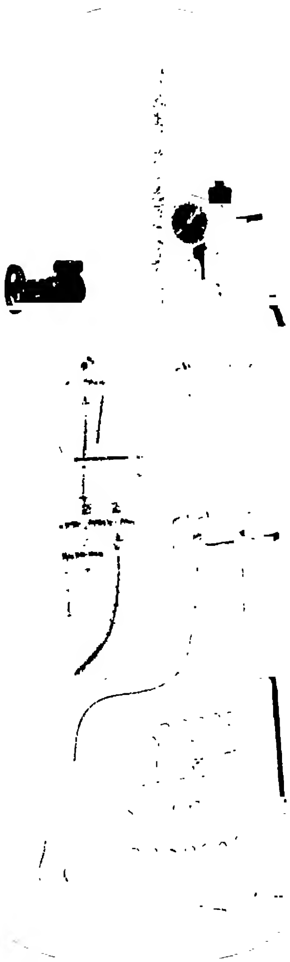


Fig. 9.—Standard Merryweather Vertical Water-tube Boiler

view of the boiler is given in fig. 9. With certain modifications the boiler can be used with sea-water for marine work. The tubes are then entirely straight, and cross one another in rows throughout the fire-box, while extra large water spaces surround the fire-box. The weight is somewhat greater in proportion than with the ordinary pattern. An especially light pattern is made for up-country work where extreme portability is essential. Here the fire-box is not jacketed, and is a separate piece on which the rest of the boiler is erected.

Fig. 10 shows the "Sentinel" boiler, as fitted to the Sentinel 6-ton steam wagon. The requirements for a boiler operating a steam wagon are, briefly, large evaporative capacity in a confined space, together with ease of firing and cleaning. This boiler is of the water-tube type, with vertical shell and fire-box. It is about 4 ft. high and 2 ft. 8 in. diameter overall, with a heating surface of 52·8 sq. ft., and 3·28 sq. ft. grate area. The boiler is built for a working pressure of 250 lb. per square inch. The outer shell is made of a single plate of boiler steel, with a vertical welded seam having a butt strap 5 in. wide. The fire-box, which is also made of a single plate, has the centre portion

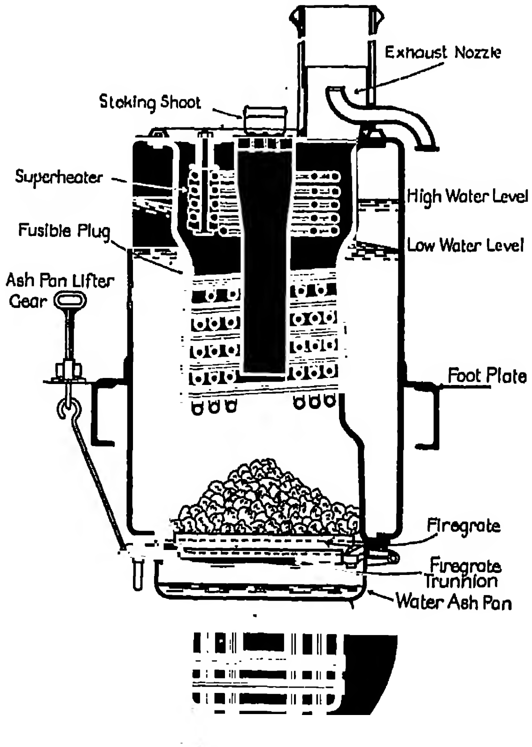


Fig. 10.—Water-tube Boiler for Sentinel Steam Wagon

squared to receive the cross water tubes. The top and bottom ends of the fire-box are flanged and riveted to the outer shell. The water tubes, which are of solid-drawn steel, are slightly inclined from the horizontal to ensure good circulation. The inner rows of tubes are screwed at each end to act as stay tubes to the fire-box. The steam passes from the steam space above the water level through a spiral superheater fixed in the top portion of the fire-box. The superheater is in this case essential, as owing to the necessarily small steam space the boiler will otherwise prime. Down the centre of the water tubes runs the fuel shoot, through which the boiler is fired. The boiler is fitted with a water ashpan, and an ashpan lifting gear for cleaning purposes. To obtain sufficient draught, an exhaust nozzle supplied with steam from the engine exhaust is fitted in the uptake.

Approximate dimensions and weights of some typical vertical boilers are given in the following table:

Maker.	Type.	Height.	Diameter.	Weight.	Grate Area.	Heating Surface Area.	Pressure.
		Ft.	Ft.	Cwt.	Sq. Ft.	Sq. Ft.	Lb./Sq. In.
Cradley Boiler Company	Cross-tube	4	2	7	2·1	10	80–100
		6½	2¾	19	3·9	40	80–100
		14	5	98	16·4	220	80–100
	Multi smoke-tube	6½	2¾	18½	3·9	75	60–150
		14	5	90	16·4	470	60–150
Spencer & Co. ..	Water-tube	6¼	2¾	21	4·2	48	100
		12	5½	130	17·7	364	100
Coltman & Sons ..	Field-tube	6½	2½	15	3	45	90
		11½	5¼	80	17·7	330	90
Cradley Boiler Company	Field-tube	6½	2¾	20	3·9	70	—
		13½	6	118	23·5	410	—
Cochran & Co.	Standard	6¾	3	23	4·75	60	100
		17	8½	255	41	1000	100
Merry-weather & Sons	Standard	3½	1¾	2½	—	—	120
		9¼	5¼	60	—	—	120

CHAPTER II

Horizontal Cylindrical Boilers

(Smoke-tube Type)

Smoke-tube boilers comprise those in which the furnace gases pass inside tubes which are surrounded by water. This type of boiler may be either internally or externally fired. In internally fired boilers the furnace

is part and parcel of the boiler, while in external firing the furnace is separate from the boiler proper.

Before describing the individual types, some general discussion of the leading characteristics will be attempted. For a steady continuous load, with not too high steam pressure, especially if plenty of floor space is available, the Lancashire and Cornish types of boiler have proved themselves efficient and reliable. Owing to the large water content and steam space, they possess a good power reserve, and fluctuation of load can take place without excessive effect on the steam pressure. The large water content also makes it possible to use hard water, as the construction renders inspection and cleaning a comparatively easy matter. The circulation of the water is sluggish, and steam cannot, therefore, be raised quickly from cold water. Steam raising must not be hastened too much, or large stresses, due to unequal expansion, will be produced; also working pressure and evaporative capacity is limited by questions of weight and dimensions. Owing to their comparatively large size and weight, and the fact that they cannot be divided into sections, the question of transport is an important item. The call for large evaporative capacity and efficiency has resulted in the production of the multitubular type of smoke-tube boiler, which has, from the nature of its construction, a greater heating surface area with comparatively smaller dimensions and weight. This type also requires in general a smaller brickwork setting, but owing to the introduction of a number of small tubes the water should be soft.

In an internally fired boiler the grate area is limited by the dimensions of the furnace tubes. For example, for a Lancashire boiler of 9 ft. diameter the greatest possible diameter of the furnace tubes is about 4 ft. In the case of hand firing, the length is also limited to about 7 ft., otherwise there will be considerable difficulty in stoking. An external furnace imposes no such restrictions, hence the grate area may be varied independently of the boiler proper, within reasonable limits. By this means the successful use of poor fuels is made possible. At the same time, the absence of furnace tubes within the boiler effects a certain simplification in construction. Several types of plain cylindrical externally-fired boilers, with the furnace directly under the boiler, have been designed from time to time; but, with the exception of small sizes, they are not in great use at the present day, the disadvantage being that in order to obtain sufficient heating surface the dimensions must be large. The local action of the fire causes unequal expansion, and therefore considerable stresses in the material, while the necessary great thickness of the material of the shell reduces the rate of heat transmission. Under-fired smoke-tube boilers are, therefore, generally of the multitubular type, where the increased heating area obtained by the tubes, and the absence of furnace tubes, makes possible a large comparative reduction in size, and so obviates the disadvantages just mentioned. Where inferior fuels are required to be used for Cornish and allied types of boiler, a special external furnace is adopted, the furnace gases being led to what normally would be the furnace tube of the boiler. Types of the multi-

tubular smoke-tube boiler designed for special purposes are the Scotch marine and locomotive boilers, which will be discussed fully later.

The Lancashire Boiler.—The construction of a Lancashire boiler is outlined in fig. 11. It consists of a cylindrical shell A, built up of several rings made of steel plate bent into cylindrical form and riveted longitudinally. These rings are connected together by circumferential riveted lap joints. As the tendency for a thin cylindrical shell to burst along a longitudinal seam is twice that along a circumference, the longitudinal joints are made much stronger than the circumferential ones. A single end plate is connected to each end of the shell. The back end plate B is generally flanged and riveted to the shell, the flange fitting inside the shell. The front end plate is connected to the shell by means of an angle ring C. In order that the flat end plates will withstand the internal pressure, they are stayed to the shell by gusset stays D. Two furnace tubes are connected to the two end plates. In the front part of each furnace tube a furnace is arranged. This consists of a set of fire-bars, arranged with an air space between each bar, and supported by a dead plate F, cross bars, and bridge plate G. In order to ensure proper draught and to prevent the fuel falling over the end of the furnace, a firebrick bridge is built up on the end of the bridge plate. The air necessary for combustion passes in at the front under the fire-bars, and so through the air spaces between the fire-bars and over the bridge. The fronts

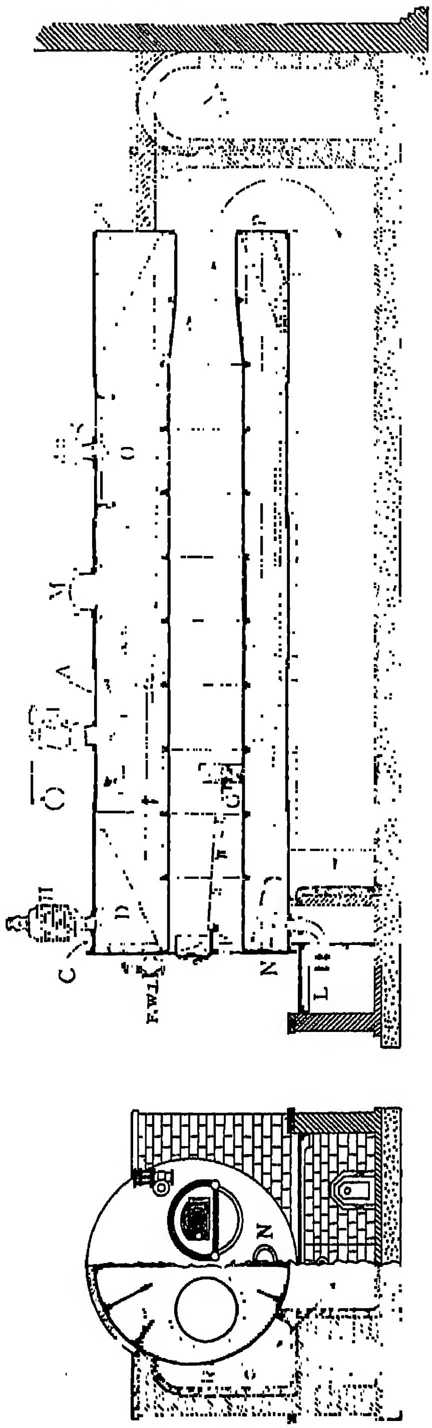


Fig. 11.—Lancashire Boiler

of the furnace tubes are closed by furnace doors above the line of fire-bars, as shown in the front elevation of fig. 11. The space in the furnace tube below the fire-bars constitutes the ashpit, and is closed below the bridge by a door. The boiler shell is fitted, by means of mounting blocks, with a deadweight safety valve H, a low-water and high-steam safety valve J, and stop valve K. The steam is dried by passing through an anti-priming pipe O, which consists of a tube closed at both ends and provided with holes along its upper surface. A manhole M and mudhole N are fitted in the shell for cleaning purposes, while by means of a blow-off cock L, sediment may be blown out or the boiler emptied. The boiler is built into a brickwork setting, with flues arranged as shown in the figure. The furnace gases, after leaving the furnace through the space between the bridge and the top surface of the furnace tube, pass along the furnace tube

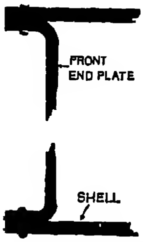


Fig. 12.—Method of Fixing Front End Plate

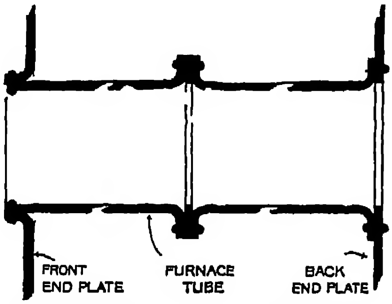


Fig. 13.—Method of Fitting Furnace Tube to End Plates

to the back, and via the downtake P along the bottom flue below the boiler from back to front. To prevent any baffling of the draught as the gases emerge from the two furnace tubes into the downtake, it is generally divided into two compartments by a central wall, which also enters the bottom flue for a foot or so. Here the gases divide into two streams which flow along the side flues R from front to back, and so to the chimney.

In some makes of boiler, especially designed for export, the front end plate is flanged and connected to the shell, as shown in fig. 12.

This method provides ease of riveting, and also has the advantage that the rivet heads are not in contact with the water; to ensure a good joint, however, very accurate fitting is necessary, and there is also a tendency for grooving to take place at the radius of the flange.

The method of attachment of the furnace tubes to the end plates varies with different makers. A common method is shown in fig. 13, where the front end plate is flanged outwards to receive the furnace tube, and the back end of the furnace tube is flanged as shown. It is essential that in the front connection the tube is an accurate fit in the flanged hole. The back connection shown is efficient if well constructed, but there is a tendency for grooving to develop. The holes in the end plates are sometimes flanged inwards, the tubes fitting in these flanges and being riveted to them. This

method, however, is not so convenient for riveting, and suffers from the disadvantage that the rivet heads are exposed to the flame and furnace gases. Connection by angle rings is a simple method, but here again the rivet heads are exposed to the furnace gases, and very accurate fitting is necessary to obviate leaks.

In order that the stresses, set up by changes in length of these furnace tubes due to variation in temperature, can be relieved, it is essential that the tubes be somewhat flexible. This is often obtained by means of " Adamson joints ", shown in fig. 13. The furnace tube is made up of short lengths, each length having its ends flanged outwards. Between each pair of flanges a caulking ring is placed, the whole being riveted together. These flanged joints give a certain amount of flexibility to the furnace tube along its axis. The rings also prevent the furnace tubes from collapsing inwards, from the boiler pressure which is acting externally on them. The design of this type of boiler is one of obtaining just the correct amount of flexibility in the various parts in order to take up expansion due to changes in temperature. Gusset stays must not make the corners too stiff, otherwise grooving and leaking will occur. For this reason the stays do not run right up to the corners but allow breathing space. The end plates should also be free to bulge slightly near the furnace tubes, for not only does the furnace tube vary in length as a whole, but owing to the top being in general hotter than the bottom half, there will be a tendency for it to curve upwards, called hogging. The construction of furnace tubes used by Messrs. Daniel Adamson & Co. is shown in fig. 68. Behind the furnace bridge, the furnace tube is made up with alternate rings of somewhat smaller diameter. This device not only gives flexibility to the furnace tube but also gives easier access to the water space between it and the shell for inspection purposes

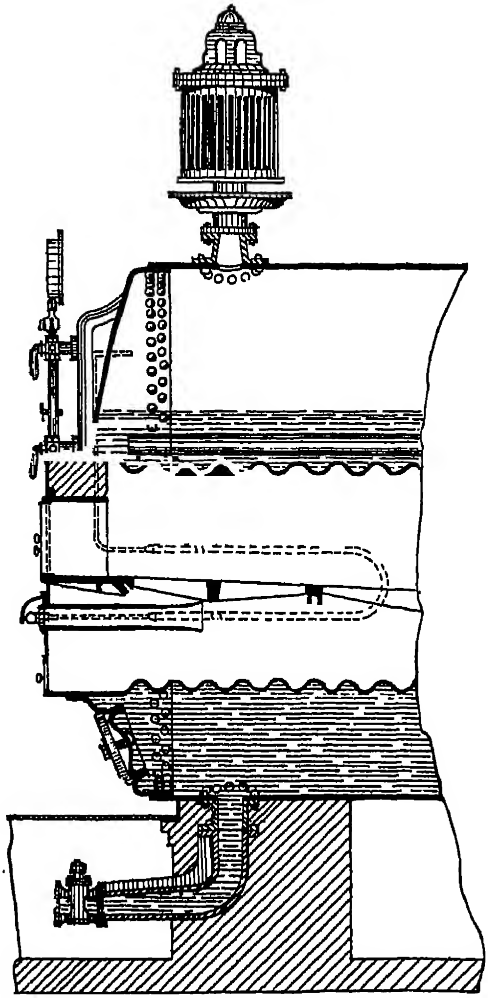


Fig. 14.—End Section of Dish-ended Boiler with Forced-draught Furnace

In order to obviate the use of gusset stays and so reduce weight and amount of riveting, the end plates are sometimes dished. They are then able to withstand the internal pressure without stays. The furnace tubes are also occasionally corrugated to allow for expansion, as shown in fig. 14, which indicates the construction of a "Thompson dish-ended boiler". Among the advantages claimed for this type are, reduced chances of leakage, ease of cleaning, due to absence of gusset stays and rivets, and increased heating surface of corrugated flues. The dished ends are, of course, owing to their construction very rigid, and grooving may occur at the flanges. Patching of the corrugated flues is not very practicable.

The essential feature of all modern boiler setting is accessibility for inspection and cleaning purposes to all parts of the boiler surface. The flues must be large enough for a man to pass down, and should therefore be nowhere less than a foot wide, while the width of the bottom flue is generally about half the width of the boiler. The amount of brickwork actually in contact with the boiler shell must be a minimum consistent with satisfactory support, as serious corrosion often occurs undetected under brickwork. It must be remembered, however, that any attempt to obtain line contact between the shell and the setting on which it rests is throwing the whole weight of the boiler on a very small portion of its surface. In order that any slight leak may be easily detected, it is important that riveted seams should not be covered. The highest point of the flues should be below the water level in the boiler to obviate any risk of overheating the portion of the boiler not in contact with water. In order to prevent loss of heat into the brickwork and foundations, an air space is sometimes provided between the flue linings and the main brickwork. Whether there is much advantage in this is an open question, for although air is a poor conductor of heat yet radiation can take place readily.

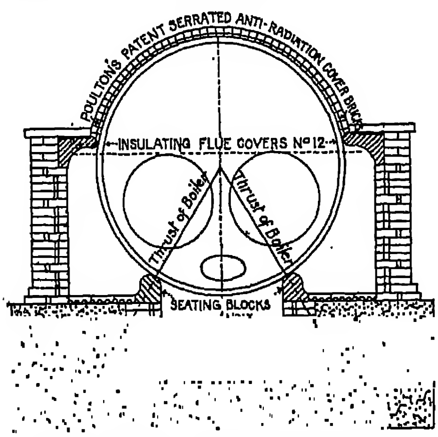


Fig. 15.—Cross-section of Setting for Lancashire Boiler

A modern boiler setting by Messrs. Poulton & Sons is shown in fig. 15. All the blocks in contact with the boiler have curved surfaces, thus giving a minimum of contact between the shell and the brickwork. The shapes of the various bricks are such as to render complete inspection possible, and there are no concealed surfaces where damp can lodge and lead to corrosion. The seating blocks and flue covers are made "long" and "short" to fit the in and out laps of the boiler plates. The flue covers are complete in themselves, thus dispensing with extra brickwork. The top surface of the boiler shell is covered with serrated cover bricks which form a stagnant

air cavity between them and the hot boiler surface. By using serrated bricks for all pavings and vertical walls of the flues it is claimed that loss of heat into the outside brickwork and foundations is reduced.

Fig. 16 shows the system of Messrs. E. J. & J. Pearson, Ltd. The boiler rests on asbestos-cushioned seating blocks. This gives a minimum of brickwork in contact with the boiler, and ensures a good tight joint, preventing gas leakage between the flues. The side flue covers can also be asbestos-cushioned as shown, or of an angular contact pattern to give the absolute minimum of brickwork in contact with the boiler. Special stepped seating blocks are provided at the seams of the shell. These blocks are made in two halves, the top section being provided with a hand hole so that it can be easily removed for inspection purposes. The top portion of the shell is covered with angular contact covering slabs made of firebrick. These bricks, which are held together by a special joint, rest on ribs which run longitudinally the whole length of the boiler. The loss of heat by radiation is thus reduced to a minimum, whilst the bricks can be readily removed and replaced without damage.

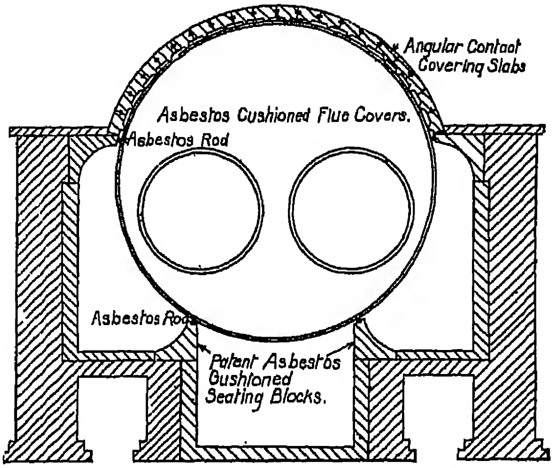


Fig. 16.—Boiler Setting with Asbestos-cushioned Seating Blocks and Asbestos-cushioned Flue Covers

The following table gives approximate weights and dimensions of typical Lancashire boilers:

Maker.	Length.	Diameter.	Flue Diameter.	Weight.	Grate Area.	Heating Surface Area.	Pressure.
	Ft.	Ft.	Ft.	Cwt.	Sq. Ft.	Sq. Ft.	Lb./Sq. In.
Coltman & Sons	18	6	$2\frac{1}{4}$	140	16	422	100
	30	8	$3\frac{1}{6}$	325	38	965	100
	18	6	$2\frac{1}{4}$	155	16	422	135
	30	8	$3\frac{1}{6}$	375	38	965	135
Ruston & Hornsby	19	$5\frac{1}{2}$	2	253	—	380	210
	19	$5\frac{1}{2}$	2	129	—	380	100
	30	$9\frac{1}{2}$	4	472	—	1159	100
	30	$9\frac{1}{2}$	4	838	—	1159	210

The Cornish Boiler.—The Cornish boiler (fig. 17) resembles the Lancashire in all respects, save that it has only one furnace tube. Owing

to the smaller grate area and heating surface, the evaporative capacity is in general less than that of a Lancashire boiler of similar dimensions. It is therefore used where a smaller rate of evaporation is required than can well be maintained with the Lancashire type. With this exception, the same remarks apply as with the two-flue type.

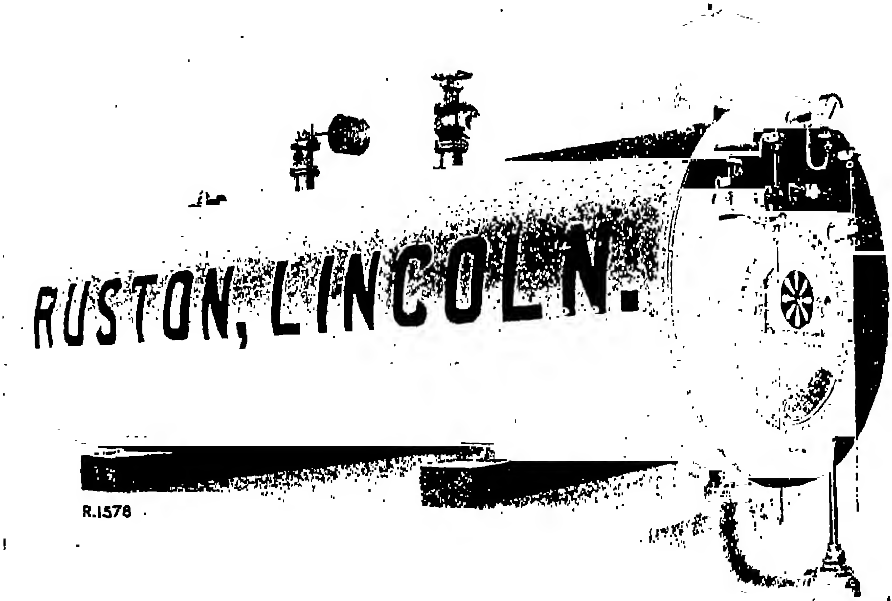


Fig. 17.—Cornish Boiler

The following table gives approximate weights and dimensions of typical Cornish boilers:

Maker.	Length.	Diameter.	Flue Diameter.	Weight.	Grate Area.	Heating Surface Area.	Pressure.
	Ft.	Ft.	Ft.	Cwt.	Sq. Ft.	Sq. Ft.	Lb./Sq. In.
Coltman & Sons	9	$3\frac{1}{2}$	$1\frac{3}{4}$	30	4·3	103	80
	24	$6\frac{1}{2}$	$3\frac{1}{4}$	165	15·2	504	80
	14	5	$2\frac{1}{2}$	80	8·7	231	100
	24	$6\frac{1}{2}$	$3\frac{1}{4}$	175	16·2	504	100
	14	5	$2\frac{1}{2}$	90	8·7	231	135
	24	$6\frac{1}{2}$	$3\frac{1}{4}$	220	16·2	504	135
Ruston & Hornsby	9	$3\frac{1}{2}$	2	53	—	99	160
	24	$6\frac{1}{2}$	$3\frac{1}{2}$	273	—	546	160

The Galloway Boiler.—In this boiler (fig. 18) the two furnace tubes are united a short distance behind the furnace into one tube of oval cross-section, which runs through to the back of the boiler, and is fitted with a

number of "Galloway" tubes. These tubes are conical in shape, and have such a taper that the smaller flange at the lower end will pass through the hole made in the upper surface of the oval flue for the large end of the tube. The introduction of these tubes, although multiplying the number of riveted joints to a considerable extent, greatly increases the heating surface area, while it is also claimed that they facilitate the water circulation, as the water on being heated can pass up the insides from the bottom of the boiler. Pockets are formed along the sides of the oval flue as shown in the figure.

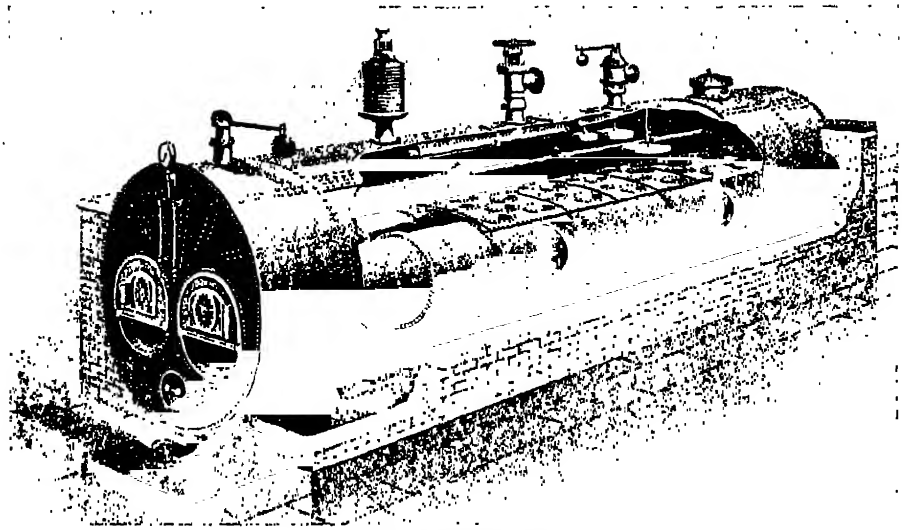


Fig. 18.—Galloway Boiler

These pockets, besides increasing the heating surface, give the flexibility to the oval flue necessary to allow for unequal expansion due to changes in temperature. The setting is similar to that of the Cornish and Lancashire boilers. The normal working pressure is from 120 to 160 lb. per square inch. "Galloway" tubes are also often fitted in the rear ends of the furnace tubes of the ordinary Cornish and Lancashire boilers.

The following table gives particulars of some standard Galloway boilers to a basis of 8·5 lb. of water evaporated per pound of coal, the boiler being supplied with a hot feed from an economizer.

Size (Feet).		Weight (Tons).		Number of Tubes.	Heating Surface Area. Sq. Ft.	Grate Area. Sq. Ft.	Steam Evaporated (Pounds per Hour).	
Length.	Diameter.	120 lb. Pressure.	160 lb. Pressure.				20 lb. Coal per Sq. Ft. Grate per Hour.	30 lb. Coal per Sq. Ft. Grate per Hour.
14	5½	7	—	6	300	14·5	2500	3,700
20	6	9·5	—	11	495	22·5	3800	5,700
30	7½	19	23·5	25	1000	35	5900	8,900
30	9	25·5	31·5	25	1310	43	7300	11,000

Multitubular Cornish and Lancashire Boilers.—In this type of boiler the heating surface is considerably increased by substituting for about half the ordinary furnace tube a number of smoke tubes of about 3 in. diameter, as shown in fig. 19, which illustrates in section a Cornish multitubular boiler made by Messrs. H. Coltman & Sons.

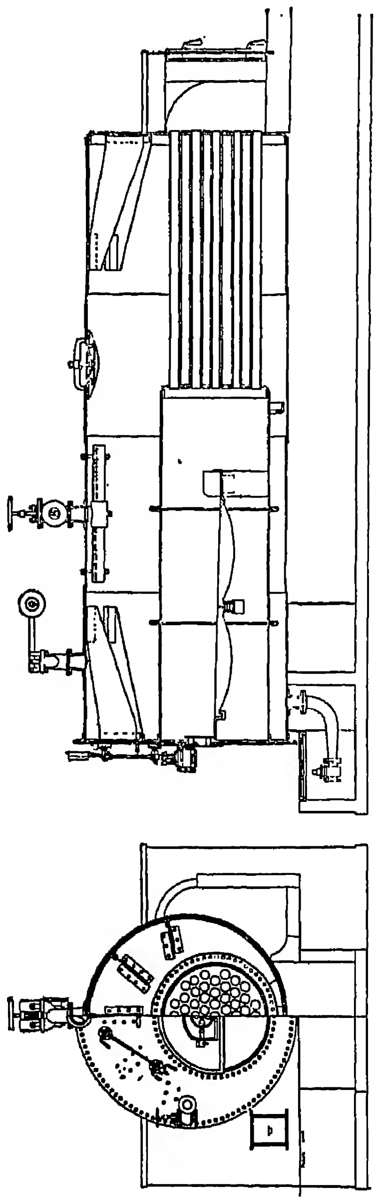


Fig. 19.—Cornish Multitubular Boiler

These boilers are usually used for pressures of 100 to 150 lb. per square inch, the furnace often being corrugated for the higher pressures. The furnace tube is sometimes placed eccentrically to the shell, to give easier access inside the boiler for cleaning and inspection. In this type it is obviously necessary to provide easy access to the back of the boiler in order to clean out the smoke tubes. Although of greater prime cost, yet compared with a Cornish boiler this type will effect a considerable saving in fuel, while for equal evaporative capacity the external dimensions are not so great. Owing to the introduction of tubes, however, cleaning on the water side is not so easy, and therefore the water must be soft to obviate the formation of incrustation, which would cause loss of efficiency, and possibly overheating of the tubes and consequent leakage. The setting of the boiler is generally similar to that of the ordinary Cornish or Lancashire type, except that, in order to facilitate access to the back of the boiler, the furnace gases pass from back to front along the side flues and back again through the central flue under the boiler, and so to the chimney. In some cases, especially for smaller sizes, only one large flue is used, the furnace gases passing off to the chimney at the front end of the boiler.

Multitubular boilers of the Lancashire type are similarly constructed, the two furnace tubes being combined about midway along the boiler into one combustion chamber, from which the gases pass into the smaller smoke tubes

Typical dimensions and weights are given in the following table:

Length.	Diameter.	Weight.	Grate Area.	Heating Surface Area.	Pressure.
Feet.	Feet.	Cwt.	Sq. Ft.	Sq. Ft.	Lb./Sq. In.
10	4	47	7	220	100
15	5	89	12·5	400	100
18	8	229	31	1000	100

A modification of this design for use in launches is shown in fig. 20. As the furnace tube is not cylindrical, it must be stayed. The flat crown of the furnace tube is supported from the top of the boiler shell by means

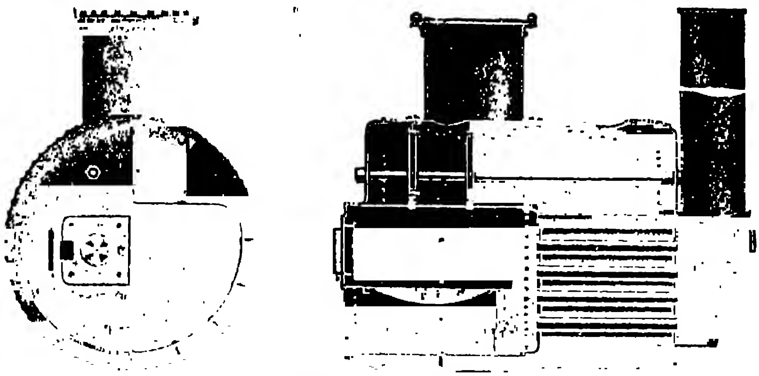


Fig. 20.—Launch Boiler

of link stays, while the curved portion is directly stayed to the outer shell as shown. A steam-dome is fitted to prevent priming. The furnace gases, passing from the fire through the smoke tubes, pass into the smoke-box, and thence to the chimney, which is 4 to 6 ft. high. Manholes and mudholes are arranged in convenient positions. Typical dimensions for 100 lb. per square inch working pressure are given in the following table:

Maker.	Length.	Diameter.	Weight.	Grate Area.	Heating Surface Area.
	Feet.	Feet.	Cwt.	Sq. Ft.	Sq. Ft.
H. Coltman & Sons ..	2¾	2¼	12	2	37
	4½	3½	32	5½	127
	6	5	66	12	304

Internally-fired Return Tubular Boiler.—This type of boiler, sometimes called the dryback or economic boiler, is designed to give a large heating surface in comparatively small dimensions, especially in length Inspection can be carried out satisfactorily, but owing to the small smoke

tubes, the water should be free from deposit, to avoid incrustation on the outside of the tubes. The design gives a high evaporative capacity and is economical, but it is best suited to a good class of fuel. Fig. 21 represents a pattern made by Messrs. H. Coltman & Sons. The flue gases, passing from the furnace tubes into a brick combustion chamber at the back, return through a number of smaller tubes to the smoke box at the front. From here they pass by means of the side flues to the chimney. The construction of the shell in fig. 21 is on the lines of the ordinary Cornish or Lancashire boiler, but in some, where the design allows no room for gusset stays, the two end plates are connected together by means of longitudinal rods, while a certain number of the smoke tubes may be of thicker material and are screwed at the end to act also as stays. Owing to the steam space being somewhat small, a steam drum is usually advisable to procure dry steam.

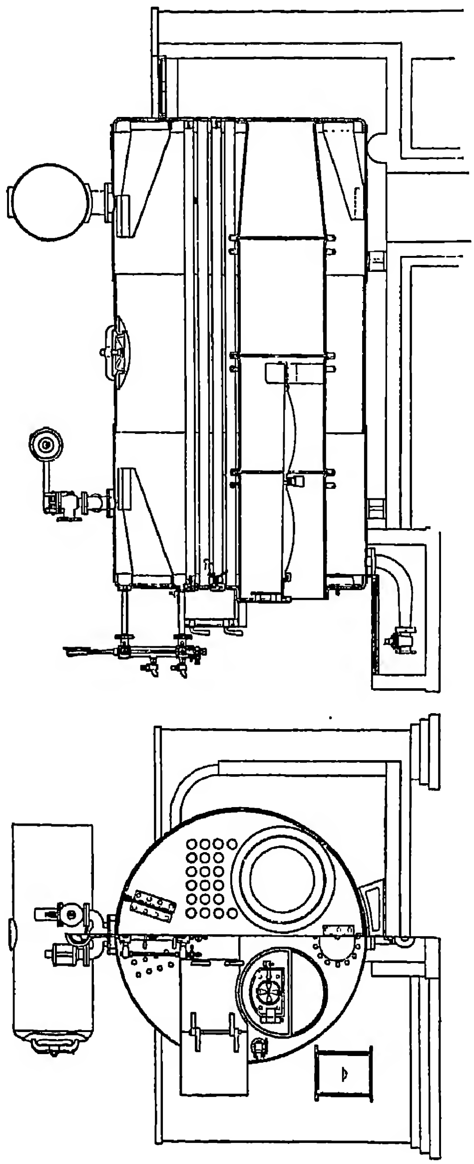


Fig. 21.—Internally-fired Return Tube Boiler

A modification of this type of boiler, in which no brickwork flues are necessary, is shown in fig. 22, which illustrates a dryback marine boiler, suitable for small vessels, made by the Farrar Boilerworks, Ltd. The boiler shell is extended at the back to form the combustion chamber, and the furnace gases, passing into this, return through the smaller tubes to the smoke-box, and thence directly to the chimney. Compared to the Scotch marine type, to be described later, there is here a certain loss of heat by radiation, but, on the other hand, this type is much cheaper to construct. The usual pressure used in this pattern is about 100 lb. per square inch. Typical dimensions are given on p. 255.

Maker.	Type.	Length.	Diameter.	Weight.	Grate Area.	Heating Surface Area.	Pressure.
		Feet.	Feet.	Cwt	Sq. Ft.	Sq. Ft.	Lb./Sq. In.
H. Coltman & Sons	1-Flue.	5	4¼	39	6·3	175	100
	2-Flue.	10	8	178	35	1062	100
Davey, Paxman, & Co.	Economic 1-flue.	6½	4¾	45	—	140	120
	Economic 2-flue.	15½	9¾	410	—	2015	120
Farrar Boiler-works	Dryback marine.	4½	3½	26	4·3	70	100
		7	5¼	68	11·3	270	100

Horizontal Multitubular Boiler (under-fired).—This type of boiler, being without furnace tubes, has small dimensions for a compara-

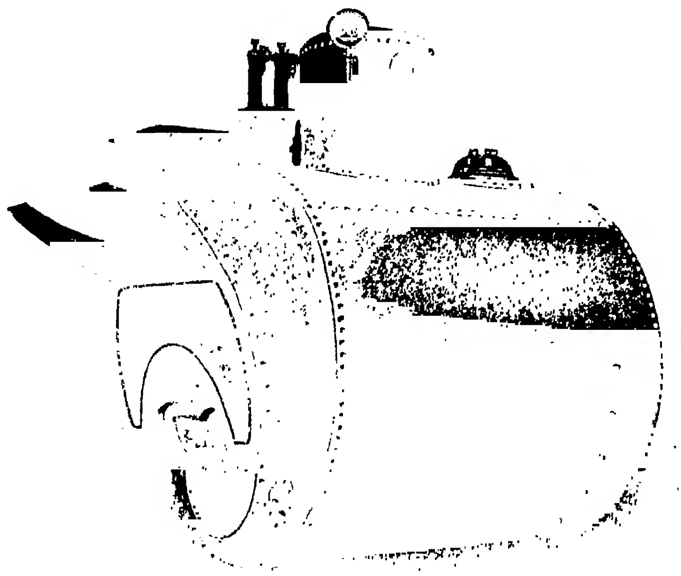


Fig. 22.—Dryback Marine Boiler

tively large evaporative capacity, and is, therefore, popular where convenience of shipment and transport are important items. The furnace being external can be varied in size within fairly large limits, and therefore can be adapted for burning inferior fuels. It consists (fig. 23) of a cylindrical shell with a number of smoke tubes running from end to end. Some of the tubes are sometimes thicker, and are screwed at the ends to act as stays to the flat ends, while gusset stays give additional support where necessary. Owing to the comparatively small steam space, a steam-dome is fitted from which the dry steam is taken, while the shell carries the usual fittings. The large number of tubes makes it a powerful steam generator in the case of emergency, whilst, if well designed, inspection and cleaning are reasonably easy. Good feed water is desirable, otherwise not only will trouble ensue, due to scale on the tubes, but, when at rest, any

impurities will settle on the bottom of the shell and tend to cause subsequent overheating. The setting varies somewhat in different plants, the furnace gases usually passing from the furnace under the boiler to the back, returning through the tubes into a smoke-box at the front, and finally to the chimney through the side flues. In this case the boiler is often supported independently of the brickwork by steel columns and cross girders. In fig. 23, which shows Messrs. H. Coltman & Sons' setting for boilers of diameter over 6 ft., the flue gases pass back along the bottom and sides, forward through the tubes into the smoke chamber, and then pass to the chimney through the small separate flues on the side. This method simplifies the setting to some extent in doing away with side brackets and supporting columns. Doors in the smoke-box give access to the tubes.

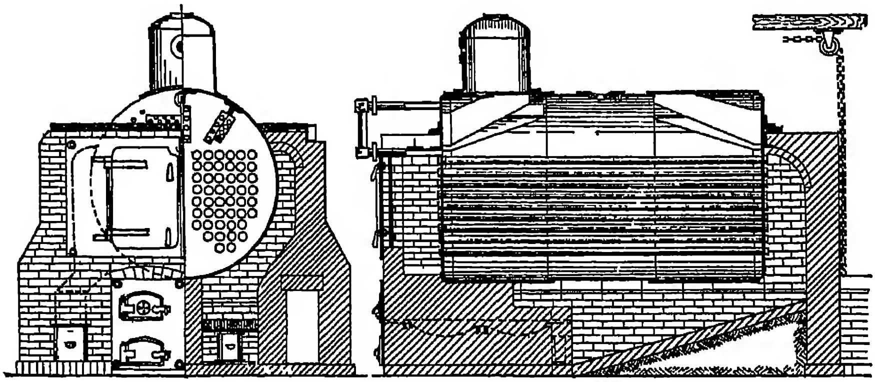


Fig. 23.—Externally-fired Multitubular Boiler

In a modified form of this boiler there are no brick side flues. The smoke box in front carries a steel chimney, the furnace gases from the smoke tubes passing straight to the chimney without returning to the back of the boiler. In this type the loss of heating surface, due to the absence of side flues, is, for some purposes, fully compensated for by the consequent simplification of the brickwork setting. Some typical dimensions are given below.

Maker.	Length.	Diameter.	Weight (Cwt.)			Grate Area.	Heating Surface Area.
	Feet.	Feet.	Pressure Lb./Sq. In. 80	100	140	Sq. Ft.	Sq. Ft.
H. Coltman & Sons.. ..	7	3½		26		5	200
	10	5½		89		16·3	650
	14	8		213		43·7	1750
Farrar Boiler-Works ..	9½	4		50		14	320
	18	6½		235		34·5	1450
Davey, Paxman, & Co.	9	4	37	41	49	—	312
	14	7	168	180	215	—	1263

Multitubular Marine Boiler (Scotch Marine Type).—Up to recent years this type of boiler was almost exclusively used in the mercantile marine, and although somewhat superseded in modern practice by the water-tube boiler, it is still used to a considerable extent. Its design affords a large heating surface without the necessity for brickwork, while it is a compact and powerful steam generator. With the generally uniform load called for in the mercantile marine, it has proved itself an efficient and economical boiler, an efficiency of 82 per cent having been obtained. It is fairly accessible for cleaning and inspection, and running repairs are reasonably simple. Heavy repairs are, however, not so simple a matter, and the removal of a boiler on board is a costly proceeding. If the ordinary

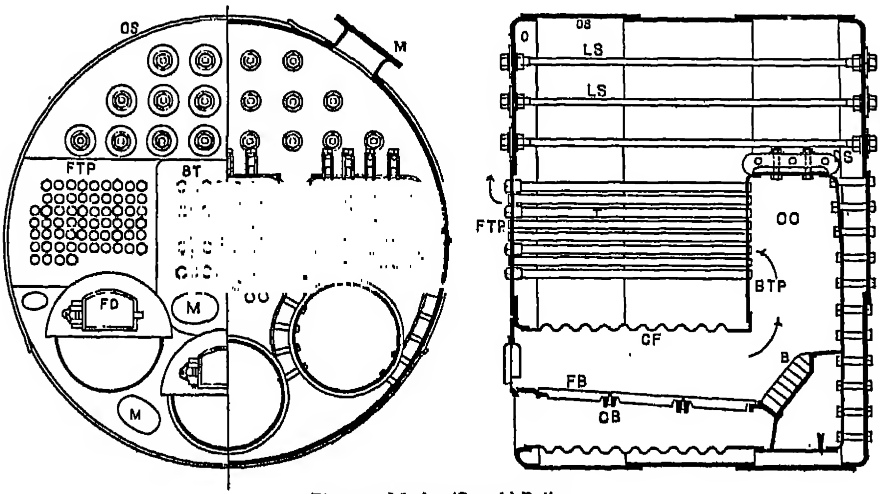


Fig. 24.—Marine (Scotch) Boiler

precaution of periodic blowing out is taken, sea water may safely be used, although the actual make-up water should be pure. Owing to slow circulation, inseparable with this design, steam should not be raised too quickly from cold water, or large stress and consequent leaks will be produced. It can be made to very large sizes, a diameter of 18 ft. being common. It is short compared to the diameter, a boiler 15 ft. diameter being about 10 ft. long. A common pressure is 180 lb. per square inch.

In fig. 24, which shows the construction of this type, a cylindrical shell OS is fitted with three corrugated furnace tubes CF, each connected at the front end to the front end plate of the shell, and at the back to a combustion chamber CC. A large number of horizontal smoke tubes T are fixed between the front end plate of the shell and the front plates of the three combustion chambers. Some of these tubes are screwed at the end and act as stays, the rest being expanded into the plates. The back plates of the combustion chambers are stayed to the back end plate of the shell. They are also stayed at the sides to one another and to the shell, while girder stays DS support the flat roofs. Longitudinal stay rods LS connect the upper por-

tion of the two end plates of the shell. The furnace tubes carry fire-bars and brick bridges, as shown in the figure. The products of combustion pass into the combustion chambers, where combustion is more or less completed, then through the smoke tubes into a smoke-box at the front of the boiler (not shown in the figure) and thence to the funnel. The furnace tubes, combustion chambers, and fire tubes, all being surrounded by water, provide a large heating surface area. Wear and tear necessitate extra strength on the bottom portion of the combustion chambers. For this reason, the bottom portions of the chambers are made of thicker plates than the sides and top. The North-Eastern Marine Engineering Co. obtain this with the minimum number of joints by constructing the top, bottom, and sides of each combustion chamber out of a single wrapper plate, rolled in one piece, with a thickened portion where it forms the bottom of the combustion chamber.

A general view of a Scotch marine boiler with superheater, made by this firm, is shown in fig. 51.

In the large sizes the marine boiler is sometimes double-ended. The furnace tubes run from end to end of the boiler, opening into combustion chambers situated centrally between the two end plates, a furnace being placed at each end of the furnace tubes.

Locomotive Boiler.—The peculiar conditions governing the design of the locomotive boiler are such as to give it very distinctive features. One of the chief factors is the question of space, which is limited; also, not only is a high rate of working necessary, but a considerable range of load is required. Weight must also be kept down to certain limits. This necessarily entails considerable variation in stress throughout the boiler, and makes the question of efficient staying somewhat complicated. Rigid economy as regards fuel and cost of production must therefore be sacrificed to some extent to fulfil the conditions mentioned. Although there are many variations in details of construction, the general design approximates to one pattern in English practice. Where broad gauges and more head room are available, as in America, the consequent greater latitude has resulted in the possibility of considerable modification.

Fig. 25 shows the general design of an English locomotive boiler. It consists of a cylindrical barrel closed at the front end by a front tube plate. The barrel consists generally of two or three plates riveted together with butt or lap joints. In front of this tube plate a smoke-box carrying the funnel is situated. The rear end of the boiler is shaped to carry an internal fire-box. This fire-box is double-walled, the space between the walls forming part of the water space of the boiler. The fire-box is connected to the smoke-box by about 200 horizontal tubes of $1\frac{3}{4}$ to 2 in. diameter. These tubes being within the cylindrical part of the boiler are all surrounded by water. Situated in the bottom of the fire-box is the furnace, from which the products of combustion pass through the tubes into the smoke-box, and thence to the funnel. In order to obtain the necessary draught through these tubes a blast pipe is placed in the smoke-box vertically below the funnel, the blast pipe being

fed with exhaust steam from the engine cylinder. The blast may be varied to obtain the draught required. The front tube plate and the rear end plate of the boiler are connected together above the fire tubes by longitudinal stays

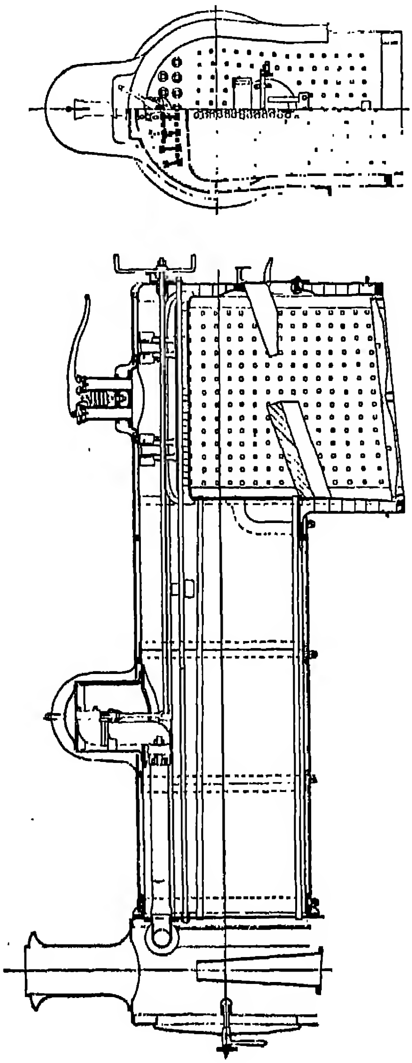


Fig. 25.—General Design of English Locomotive Boiler

about 1 in. in diameter, while across the rear end above the fire-box are sometimes similar cross-stays. The smoke tubes themselves generally give enough support between the flat surfaces they connect. The tubes, which are made of solid-drawn steel, are slightly larger in diameter for a few inches at the smoke-box end to give facility in fixing. They are expanded

into the tube plates, the ends being beaded over. Some of the tubes in the upper nests are often made about 4 in. in diameter to take superheater elements, which are described later (p. 291). On the top of the boiler is a safety valve, and a steam-dome by which priming is prevented as far as possible.

The "Belpaire" fire-box is shown in fig. 26. It consists of an inner fire-box and an outer shell. The outer fire-box consists of the wrapper plate forming the sides and top, the throat plate which forms the front, and the back plate acting as the rear end plate of the boiler and containing the fire-hole. The inner fire-box is generally of copper, although steel ones are also in use. The inner and outer fire-boxes are tied together on the flat sides and

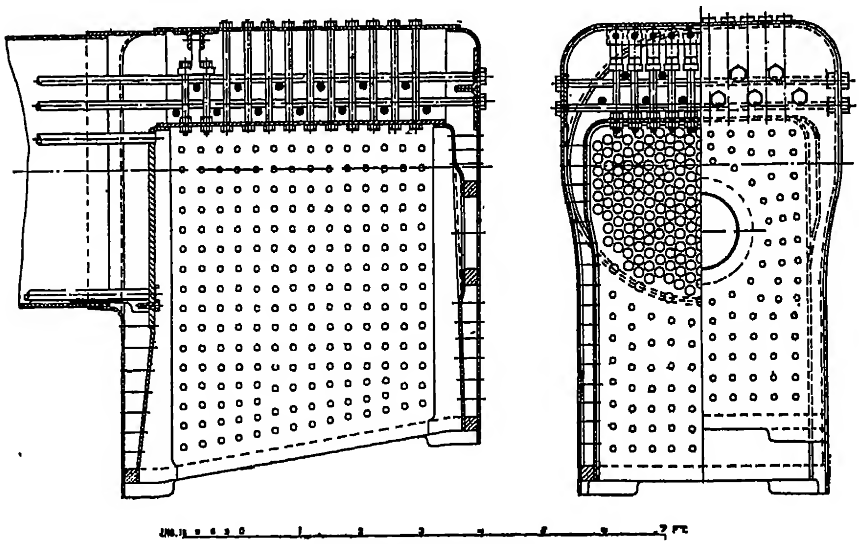


Fig. 26.—Belpaire Fire-box

also on the flat back and front surfaces by a large number of copper stays. With the fire-boxes in position, the stay holes are tapped and the threaded stays are screwed in, the ends being finally hammered over. Copper is very suitable for the fire-boxes and stays owing to its ability to withstand the varying stresses produced, and also because of its high conductivity for heat and comparative resistance to corrosion and wasting. The crown of the inner fire-box is also stayed to the top plate of the outer shell. The fact that in this pattern both the crown of the fire-box and the outside shell are flat, makes it possible for these stays to pass direct between the two plates. The first one or two stays over the front plate of the inner fire-box are of a special pattern to allow a rise of the crown of the inner fire-box but not a bulge downwards. This is to allow for the expansion of the throat plate due to its being temporarily hotter than the other parts, especially when first raising steam. The method of staying the crown of the fire-box when the outer shell of the boiler is not flat is by means of girder stays as shown in fig. 25. The crown

plate is fixed to these girder stays by means of bolts, so that it is supported independently of the outer shell, although some of the girder stays are also usually connected to the outer shell by sling stays.

Dimensions of these boilers vary with different designs. One design of a four-cylinder express engine of the Atlantic type has a heating surface area of 2352 sq. ft., with a grate area of 31 sq. ft., the working pressure being 175 lb. per square inch.

The high evaporative capacity, combined with compactness and portability,

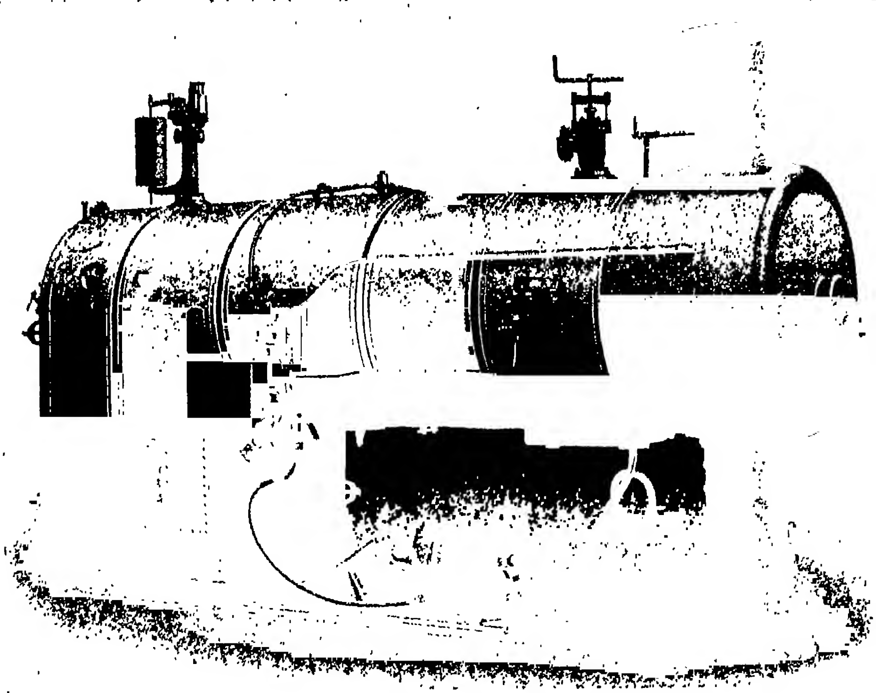


Fig. 27.—Locomotive-type Boiler

of the locomotive type of boiler, makes it very suitable for temporary stationary work. Such a type, made by Messrs. Robey & Co., is illustrated in fig. 27. The boiler barrel, fire-box, and smoke-box tube plate are of mild steel. The tubes are solid-drawn steel, expanded and beaded over at the fire-box end. The tubes project through the smoke-box tube plate, and are swelled for easy removal. The method of staying the fire-box is shown in fig. 28. The girder roof stays are carried by angles riveted to the sides of the outer fire-box shell. In this way the strains are not on the top of the boiler shell but are transferred to the outer side plates. The design also allows ample room for cleaning. The boiler is specially designed to allow it to be put on wheels if necessary, to provide easy transport. In this way the boiler as it stands is a complete steam raiser and is immediately ready for work.

Messrs. Marshall Sons & Co. dispense altogether with crown stays to

the fire-box, by using a corrugated top to the inner fire-box. The corrugations spring from the opposite corners of the fire-box, cross diagonally in the middle, and form an exceptionally strong truss to the crown plate. The shape of the corrugations enables them to breathe freely, and the absence of stays allows the top of the fire-box to be cleaned readily.

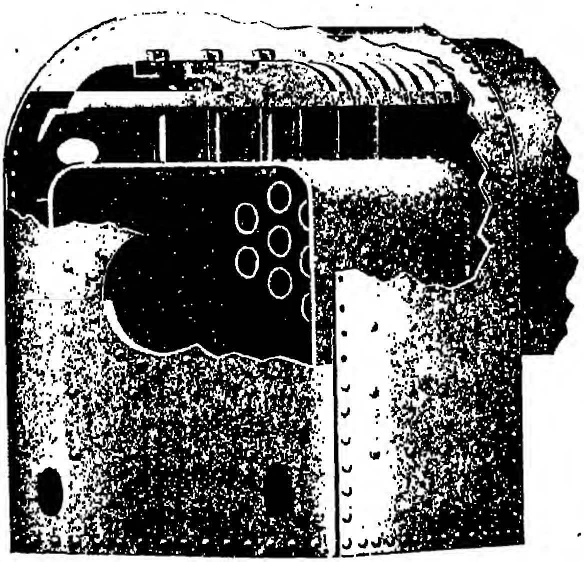


Fig. 28.—Girder Stays

Fig. 29 shows the construction of a boiler of the locomotive type, made by Messrs. Ransome, Sims, & Jefferies, for use with steam wagons. Its construction is very simple, and contains a minimum of joints. Two pressed-steel plates form the shell, the longitudinal joint being welded and strengthened by a single riveted internal butt strap. A single plate pressed into a dome shape forms the fire-box, which is inserted from below, and riveted round the base. A tube plate is fitted within the horizontal portion of the shell, the front portion of which forms the fire-box. A bank of smoke tubes is fitted between the front

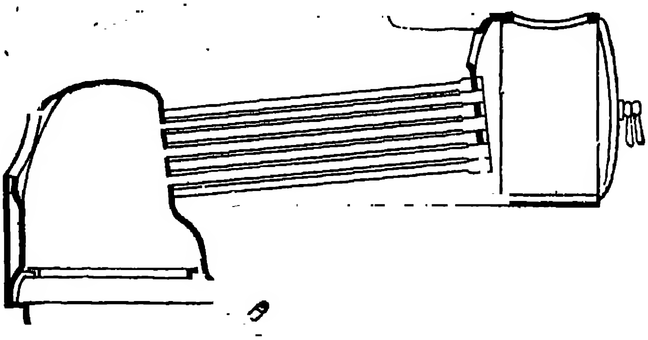


Fig. 29.—Locomotive-type Boiler for Steam Wagons

tube plate and the front side of the fire-box. It will be noticed that there are no fire-box stays, and consequently cleaning and repairs are facilitated. Four handholes, fitted with covers, are provided just above and around the foundation ring, and another is fitted on the shell over the fire-box. The boiler, which works at a pressure of 220 lb. per square inch, supplies

steam direct to the engine cylinders. Owing to consequent complication in lubrication, no superheater is used, although a feed-water heater is fitted.

CHAPTER III

Water-tube Boilers

Modern demands for high steam pressure and ability to raise steam quickly have resulted in the production of water-tube boilers on an extensive scale. In a water-tube boiler the water and steam circulate inside a series of tubes of comparatively small diameter, while the furnace gases pass round the outside of these tubes. The comparative disadvantages of the earlier designs of water-tube boilers have now been reduced to such an extent that in modern practice water-tube boilers, which are the only type used in the navy, are rapidly replacing other types in mercantile marine work, and are strong competitors to the cylindrical boiler in land practice. The case of the water-tube boiler versus the cylindrical pattern in marine practice is well presented in a paper by Mr. A. Spyer,* to which the reader is referred for detailed information.

Modern practice demands, in the perfect boiler, high pressure (consistent with safety), large steam and water storage, good circulation, economy, high and rapid evaporative capacity, possibility of using poor fuels, and accessibility, with minimum dimensions and weight. Before describing the various patterns in detail, it will be advantageous to discuss how closely the water-tube boiler approaches to these requirements.

The construction permits of the whole boiler being supported from a steel framework independently of the brick settings. There is freedom of expansion and contraction, due to changes in temperature, resulting in a possibility of forcing the boiler in an emergency without causing leakage at the joints. The furnace, being external to the boiler proper, can be varied within wide limits in the design, which permits of poor fuel being used, while the heating surface can also be of large area, enabling great evaporative capacities to be obtained in comparatively small dimensions. The sectional construction is an important item with regard to replacement and transport, while the important point of ready accessibility to all parts for inspection, cleaning, and repair, has been fairly well solved in modern designs of water-tube boilers. It should be noticed that while there is comparative saving in floor space in the case of water-tube boilers, more head room is required than in the case of Lancashire or Cornish boilers. High pressures can be obtained without undue increase in weight. While in the smoke-tube type the pressure acts externally on the furnace tubes

* "Water-tube versus Cylindrical Boilers in the Mercantile Marine", read before the Liverpool Engineering Society, 3rd March, 1920.

and smoke tubes, in the water-tube type the tubes have to withstand an internal pressure only. This is obviously an advantage, as any slight variation from a cylindrical section will tend to be augmented by external pressure but neutralized by internal pressure. The tubes in the water-tube boiler can therefore be of comparatively thin material, thus producing a saving in weight as compared to the cylindrical type, in which the necessary thickness of material becomes excessive with high pressures. The steam-and-water drums, which are separate from the water tubes, can also be of reasonable dimensions. The thinness of the water tubes is also of decided advantage in that it ensures a uniform distribution of mechanical stress.

Water capacity being fairly small and circulation large, steam can be raised quickly, while the boiler can also be cooled rapidly for repairs or cleaning. The water capacity must, however, not be too small, or for large loads there will be a difficulty in maintaining the water level and the pressure constant. In a battery of boilers fed from a common pipe this is a great disadvantage, and automatic feed regulators are necessary. Small water and steam content are obviously an advantage in the case of an explosion or burst tube, the consequences being less serious than with the Lancashire or Scotch marine type.

As the design depends on a large circulation through a number of comparatively thin-walled tubes, any formation of deposit on the inside of the tubes will restrict the circulation, and also impede the flow of heat from the furnace gases, thus causing burnt tubes. It may, however, be pointed out that Sir John I. Thornycroft found that in the "Daring" type of boiler the circulation was actually so great as to polish the insides of the tubes, thus preventing formation of deposit. Any corrosion, due to impurities in the water, is also a serious matter. The presence of carbon dioxide is one of the most common causes of corrosion, while corrosive acids are also formed by the decomposition of certain lubricating oils. It is therefore important that the feed water be pure and, as far as possible, free from air. This, however, also applies to cylindrical boilers, although the effects are not so immediately apparent in their case. External corrosion of the tubes, due to the accumulation of deposit or condensation from the furnace gases on the cooler portions of the tubes, must also be guarded against.

Babcock & Wilcox Boiler (Land Type).—Fig. 30 shows the W.I.F. type of boiler, constructed throughout of wrought steel. A number of solid-drawn steel tubes of about 4 in. diameter slope downwards from front to back. The tubes in each vertical row are connected front and back by a header. The headers are of such a form that the tubes are staggered, so that a tube in any horizontal row comes over a space in the next horizontal row. Each header is connected by a tube with the horizontal water-and-steam drum; while the bottoms of the rear headers are also connected by short tubes to the mud drum. The openings in the headers for cleaning, opposite the end of each tube, are closed by handhole

plates, as shown in fig. 31. For pressures above 200 lb. per square inch these handholes are fitted with internal caps of oval construction. By

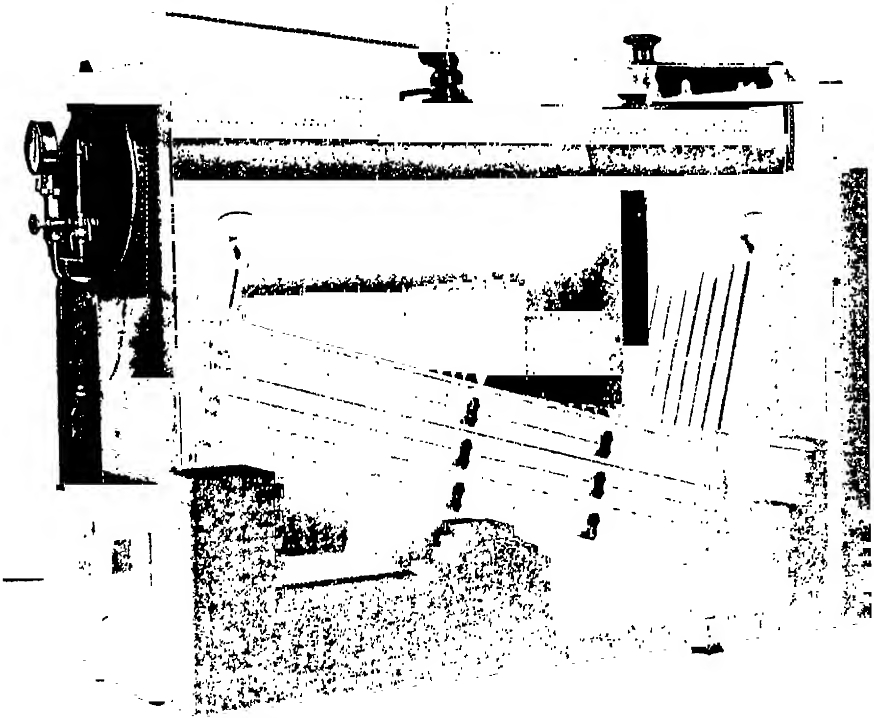


Fig. 30.—Babcock & Wilcox Boiler—W.I.F. Type

this means the greater the pressure on the cap the tighter the joint becomes, and no undue strain is placed on the tightening-up bolt.

The boiler is suspended from wrought-iron girders resting on iron columns, and is entirely independent of the brickwork, thus avoiding any straining of the boiler due to unequal expansion between the boiler parts and the brickwork, and also permitting the brickwork to be easily removed in case of repairs. The furnace is under the front end of the tubes, special attention being paid to the provision of a combustion chamber of suitable height in order that complete ignition of the gases may take place before reaching the tubes. The nest of tubes is divided by two vertical baffles built up of special tiles of particular

Fig. 31.—W.I.F. Header and Handhole for 200 lb. Pressure

construction and design, so that the gases first pass straight up between the front portion of the tubes into a chamber usually containing a superheater, and are then directed downwards into a back chamber, where any ash or soot can be easily collected, then finally up again to the rear portion of the tubes and away either to an economizer or direct to the main flue. The circulation of the water is well defined. As the water is heated it will rise towards the higher end, and the mixture of steam and water will pass from the front headers through the vertical passages into the drum, where the steam separates from the water, the latter flowing to the rear of the drum and then down the rear passages again, thus producing a continuous circulation. It will be noticed that, as the passages are large and the circulation therefore rapid, there is a thorough mixing of the water throughout the boiler, and a consequent fairly equal temperature, preventing to some extent the formation of deposits on the heated surfaces by sweeping them away and depositing them in the mud drum below the rear headers. The steam is taken from the rear end of the boiler in order that it may be thoroughly separated from the water. Approximate dimensions of a few standard sizes are given below.

Length. (over brickwork).		Width.		Height.		Weight (packed).	Heating Surface.	Furnace Area.	Evaporation.
Ft.	In.	Ft.	In.	Ft.	In.	Tons.	Sq. Ft.	Sq. Ft.	Lb. per hour.
9	6	4	5	10	5	$3\frac{3}{4}$	119	5·2	360
21	0	6	10	14	1	12	983	19·15	3000
23	0	8	7	16	8	$21\frac{1}{4}$	2010	39	6100
23	6	20	10	22	11	$80\frac{3}{4}$	9273	196	28000

The last has three steam-and-water drums joined by cross pipes.

Babcock & Wilcox Boiler (Marine Type).—This boiler is on the same lines as the land type, with certain modifications to adapt it to marine practice. Efficiency on board ship is not merely a question of water evaporated per pound of fuel consumed. Weight and dimensions are very important factors, while quick steam raising, and in the case of war vessels the possibility of overloading temporarily, are also points to be considered. It is extremely important that repairs can be carried out in the comparatively confined space on board ship by the ship's staff.

The boiler shown in fig. 32 consists of a number of inclined tubes divided into vertical sections, each section being connected by headers similar to those in the land type. The tubes are inclined, however, upwards from the front. One or more of the lowest horizontal rows of tubes are of larger diameter than the rest. Extending across the front of the boiler, and connected to the front headers by short tubes, is a horizontal steam-and-water drum. This drum is fitted with wash plates to prevent undue movement of the water when the ship rolls. The upper ends of the rear headers are also connected by horizontal tubes to this drum. Across the bottom

of the front headers, and connected thereto by short tubes, is a steel box of square section, forming a sediment box through which the boiler can be completely drained. The two ends of the steam-and-water drum are connected to this sediment box by large downcomer tubes. Opposite the end of each tube in the header is a handhole closed by a steel door and stud; by means of these holes the tubes may be cleaned or renewed. The furnace is built of ordinary firebricks, or of light fire tiles, which are bolted to the

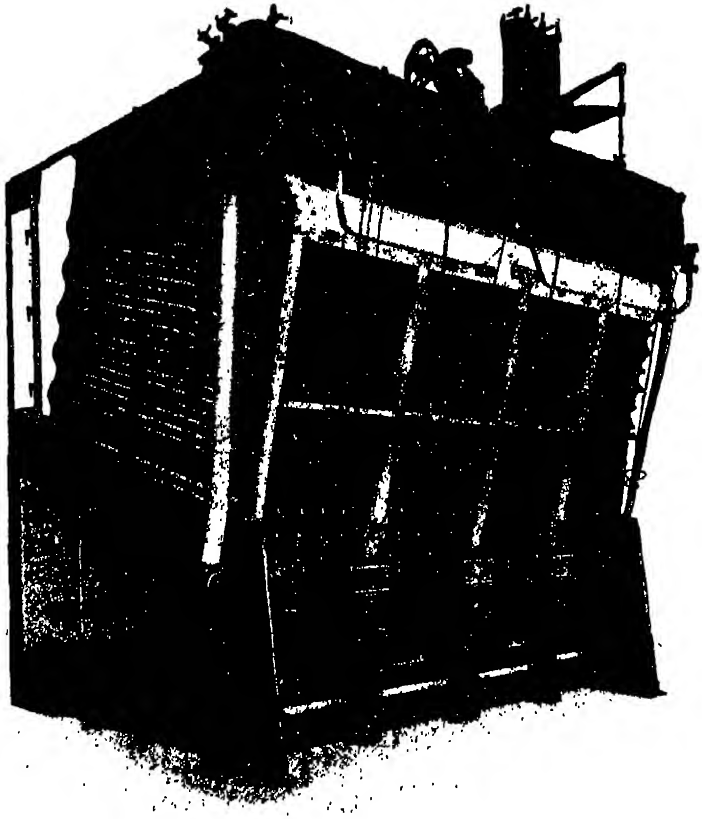


Fig. 32.—Babcock & Wilcox Marine-type Boiler

side plates, the whole being encased in a plating fitted with fire refractory non-conducting material. A roof of baffle bricks, extending from the front to about half-way towards the back of the furnace, is placed over the lower and larger inclined tubes, while a wrought-steel plate placed over the top row of inclined tubes, together with vertical baffles fixed among the tubes, guide the furnace gases along the correct paths. By this means the furnace gases pass up at the back through the tubes, then down the tubes between the two vertical baffles, and finally up through the front portions of the tubes to the funnel. The feed water enters the steam drum through an internal feed nozzle with a contracted orifice above the water line, so that

the feed water being discharged into the steam space is freed from air. The water from the drum passes down the front headers, and, being heated in the inclined tubes, rises with the steam formed to the rear high end. It then flows through the rear headers and horizontal return tubes to the steam and water drum. Upon entering the drum the mixture of steam and water impinges against baffle plates, which causes the water to be thrown downwards, while the steam passes round the ends of the baffle plates to the steam space, from whence it is taken through an anti-priming pipe to the stop valve.

Babcock & Wilcox Cross Type Marine (C.T.M.) Boiler for Land Purposes.—Fig. 77 shows a cross-section of this boiler. It has been particularly designed to deal with the high evaporative capacities demanded by the modern electric power station, and has been constructed up to pressures of 500 lb. per square inch, and to give evaporations in a single unit of over 100,000 lb. of steam per hour.

In construction the boiler closely follows the lines of the B. & W Marine Type, inasmuch as the whole length of the bottom tubes is exposed to the furnace, and consequently a high rate of evaporation per square foot of heating surface is obtained.

A special feature of the C.T.M. unit, however, is that a steel tube economizer is superposed above the boiler, forming an integral part of the plant, the whole being totally enclosed in a steel framework and steel casing lined with refractory material, by means of which radiation losses are reduced to a minimum. A common standard size for this boiler is an evaporation of 50,000 lb. of steam per hour, although they are also made in smaller sizes, and, as has already been mentioned, have already been constructed for over double this capacity.

Owing to the superposed economizer, the evaporation of this unit per square foot of ground space occupied is very high, while owing to the compact constellation of the various heating surfaces, and the provision made by means of the steel casing to prevent radiation losses, efficiencies of over 86 per cent have been obtained

Marshall Water-tube Boiler.—This boiler is of a very simple and efficient type. It combines large water and steam storage with good circulation and evaporative capacity. It consists (fig. 33) of a cylindrical steam drum, with a steel header or water-leg riveted direct to the drum at each end. A series of straight solid-drawn water tubes connect the headers, the tubes being expanded in holes in the inner plates of the water-legs. The tubes are parallel with the drums, and the whole boiler is set in brickwork with an incline downwards to the back end. The front and back plates of the headers are stayed together by hollow steel-screwed stays, so that a steam jet can be passed through the stays to clean the exterior of the water tubes. The headers afford a large cross-sectional area of steam and waterway, and thus obviate the disadvantage which is sometimes put forward against the use of horizontally-inclined water tubes. The outer plates of the headers have holes with removable lids opposite each tube

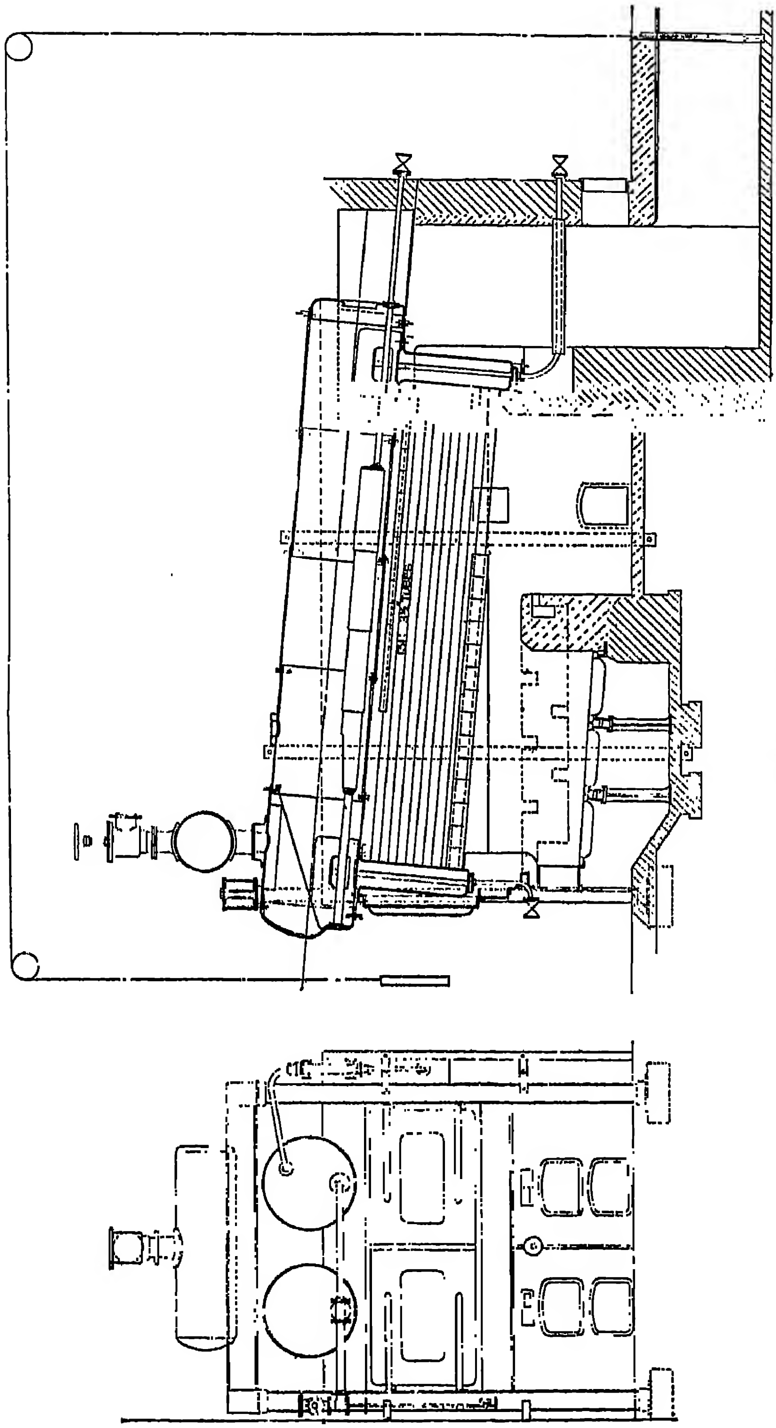


Fig. 33.—Marshall Water-tube Boiler

end, for cleaning or the removal of the tubes. The back header has a blow-off cock for removing sediment. By means of suitable baffles the furnace gases pass along the tubes from back to front, and then under the steam-and-water drum from front to back, to the chimney flue. The larger sizes have two steam-and-water drums. The boilers are specially adapted for burning saw-mill refuse and other inferior fuel, as the grate area is large. The standard boiler works at 150 lb. per square inch pressure, but can be used for pressures up to 180 lb. per square inch.

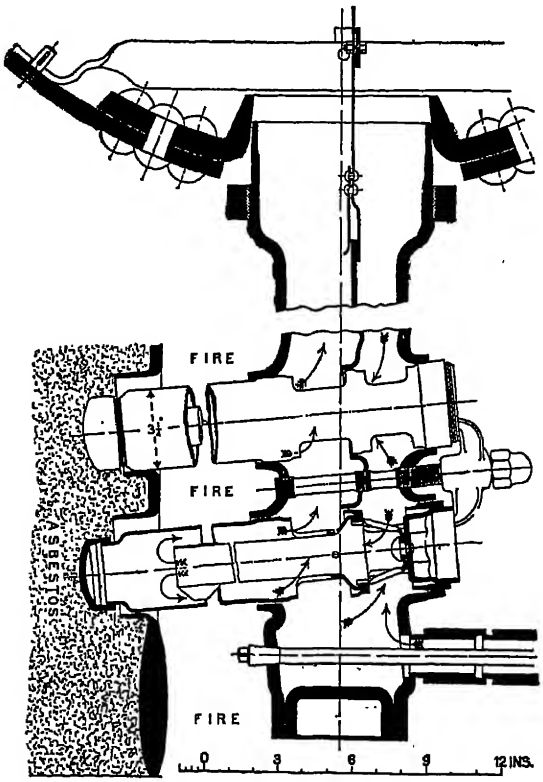


Fig. 34.—Header and Evaporative Tube Ends of Niclausse Boiler

Niclausse Boiler. — This boiler embodies many original points in its design. The heating surface consists of a large number of horizontally-inclined tubes consisting of two parts (fig. 34), an outer and an inner tube. The outer tube, which is of solid-drawn steel, has swellings on the front end, to form cones which fit into headers of solid-drawn pressed steel. The back end of the outer tube is closed by a cap and is carried by a supporting plate. The inner tube, which is in direct communication with the outer tube at its back end, is supported at the front end, which is opened out to fit into the outer tube. The water circulation in these tubes is along the inner tube from front to back, returning from back to front in the space between the inner and outer tubes. The outer tube has two machined coned surfaces, while the front and back walls of the header contain machined coned holes, into which the cones on the tubes fit. A steam- and water-tight joint is obtained by making the back cone on the tube somewhat flexible. It comes into contact with its coned hole in the back wall of the header just before the front cone, which is rigid, does so in the coned hole in the front wall of the header. The tubes are held up in position by means of bridge pieces. Each header is divided into two compartments by a vertical partition. Ports cut into the water tubes connect the insides of the inner tubes with the front compartment of the header, and the spaces between the inner and outer tubes with the rear compartment of the header. The headers are all attached

to the bottom of a cylindrical steam-and-water drum. This drum is divided into two compartments by a longitudinal diaphragm, which is in prolongation of the vertical partitions in the headers. The vertical headers are all connected at the bottom to a horizontal header, or " bottom box " (fig. 35).

In three out of every four headers there is a diaphragm E in the front compartment, which separates the top part of that compartment from the bottom. These headers are shown at A fig. 35. Every fourth header has a single front compartment as shown at B, this front compartment being continued up above the water line into the steam space in the steam-and-water drum. The feed water enters a trough C in the steam-and-water drum. Any impurities are precipitated here, and can be blown off through the valve D. The water passes down the front compartments of all the headers A to the diaphragm E. It then passes through the inner water tubes, is heated by the outer layer of hot water, and then, passing between the inner and outer tubes, the water together with any steam formed returns up the rear compartments of the headers to the water drums. This hot water can now pass right down the front compartments of the headers B. Part is converted into steam by passing through the double water tubes and then back to the steam-and-water drum; the other part flows through the bottom box to the bottom tubes of the sections A, and so back to the steam-and-water drum.

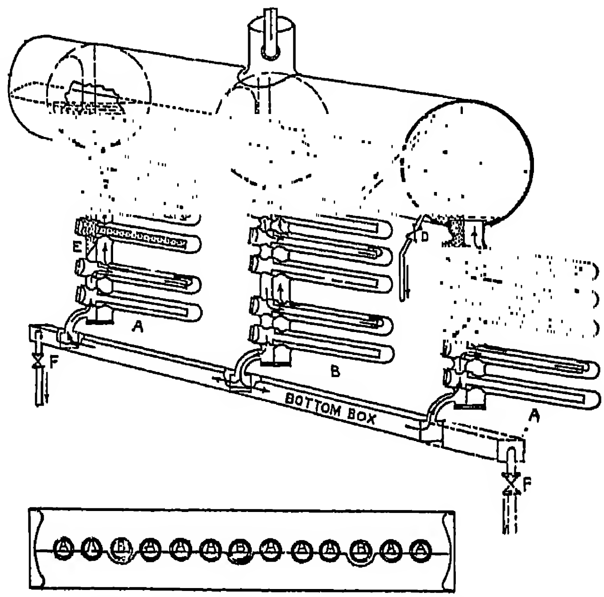


Fig. 35.—Diagram of Water Circulation in Land-type Boiler

In the marine type (fig. 36) this circulation is somewhat modified, in that all the headers have a diaphragm in the front compartment, and the hot water is fed to the tubes below these diaphragms by two vertical tubes connecting each end of the steam-and-water drum with the bottom box. By so feeding the upper tubes with relatively cold water, while the lower tubes, which are most exposed to the heat of the furnace, receive water at the steam temperature, it is claimed that any impurities are practically harmless, since the temperature of the gases surrounding the upper tubes is insufficient to form a hard scale. In order to increase the gas flow in the high-duty marine boiler having no superheater, a number of plain open-ended tubes are laid between the evaporating tubes.

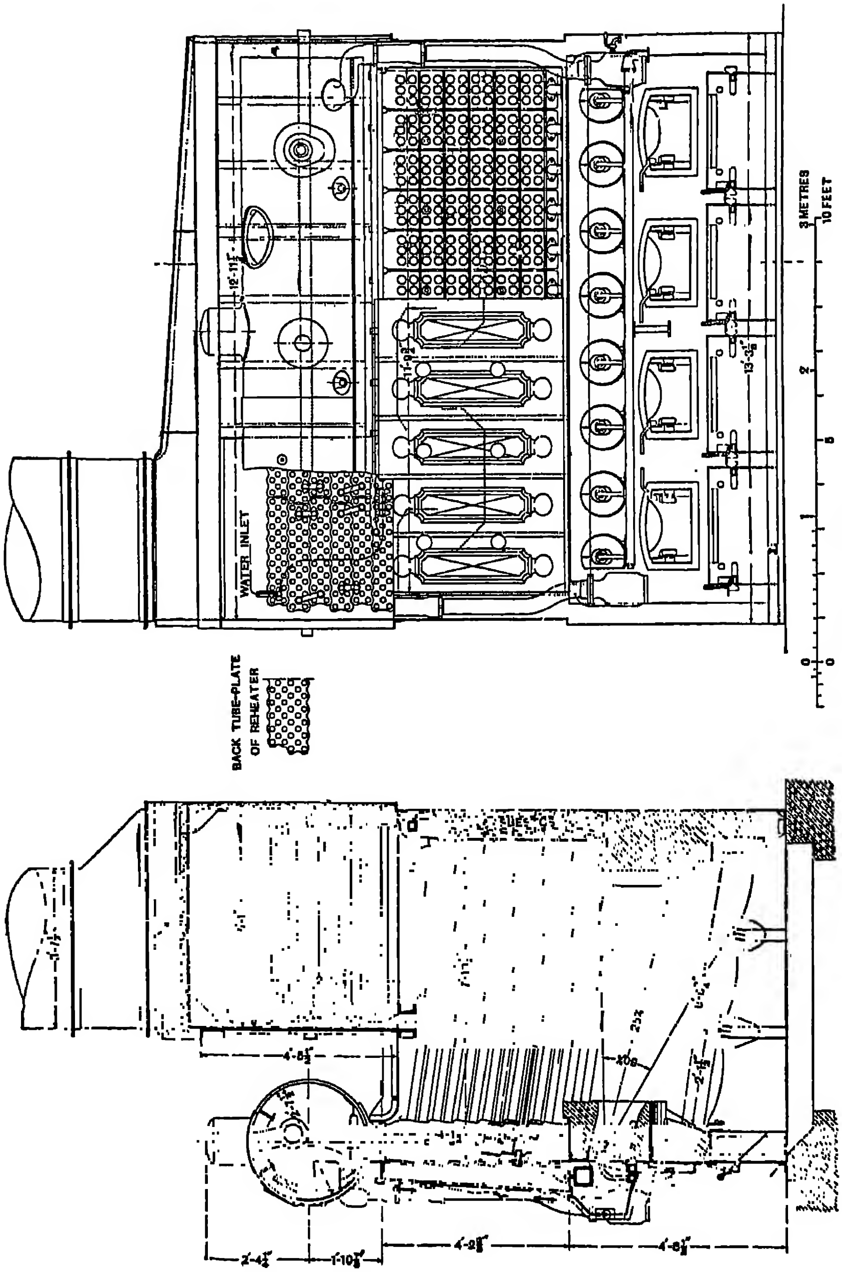


Fig. 36.—High-duty Marine-type Boiler

Tests on this boiler have shown that it has a very high thermal efficiency, a value of 91½ per cent having on one occasion been reached. This is doubtless due, in some part, to the feed water receiving a preliminary heating before passing through the lower and hotter tubes, while each individual tube acts as a feed-water heater. The boiler is low in height and, due to the com-

paratively short tubes, takes up a small ground space. It is claimed that the boiler is adaptable for any type of service. Details of trials may be found in the Proceedings of the Institution of Mechanical Engineers.*

Clarke-Chapman Boiler (Woodeson's Patents), Land Type.—This boiler consists of one or more sections, each section consisting of a horizontal cylindrical steam drum at the top, a cylindrical water drum at the bottom, the two drums being connected by several groups of vertically inclined tubes expanded into flat portions on the steam-and-water drum. Fig. 37 shows the three-section type. The water drums are connected together by horizontal tubes, while the steam drums are connected in a

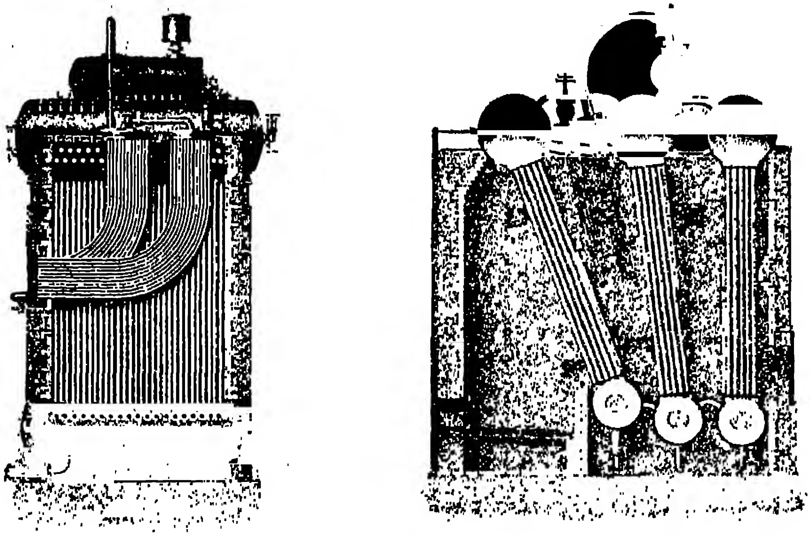


Fig. 37.—"Woodeson" Patent Boiler, Hand-fired

similar manner both below and above the water line. On the top of the steam drums are a number of manholes over each group of tubes. The fire grate is arranged across the boiler under the front section, the furnace gases being directed by baffles upwards through the tubes of the front section, down through the middle section, thence up between the two baffles, and finally down through the rear section of tubes to the chimney. The whole boiler is suspended from girders and is thus free to expand (fig. 38). The feed water enters the rear drum and flowing down the rear tubes, which are in contact with the coldest gases, precipitates any deposit into the bottom rear drum. The water then rises in the tubes of the other two sections and the resulting mixture of water and steam is separated in the steam drums, the steam passing away to the steam dome situated over the centre steam drum. (Note, fig. 38 includes a superheater between the front and centre sections.)

It will be noted that in this boiler there is a fair water capacity to prevent

* *Proc. Inst. Mech. E.*, July, 1914, p. 507.

undue fluctuation of pressure. The arrangement of the tubes allows a large combustion chamber, thus enabling combustion to be fairly complete before coming into contact with the tubes. The makers claim that this obviates the loss due to the cooling effect if the gases come into contact with the heating surface before combustion is complete. The tubes being perfectly straight facilitates inspection and cleaning, while as they are all of the same

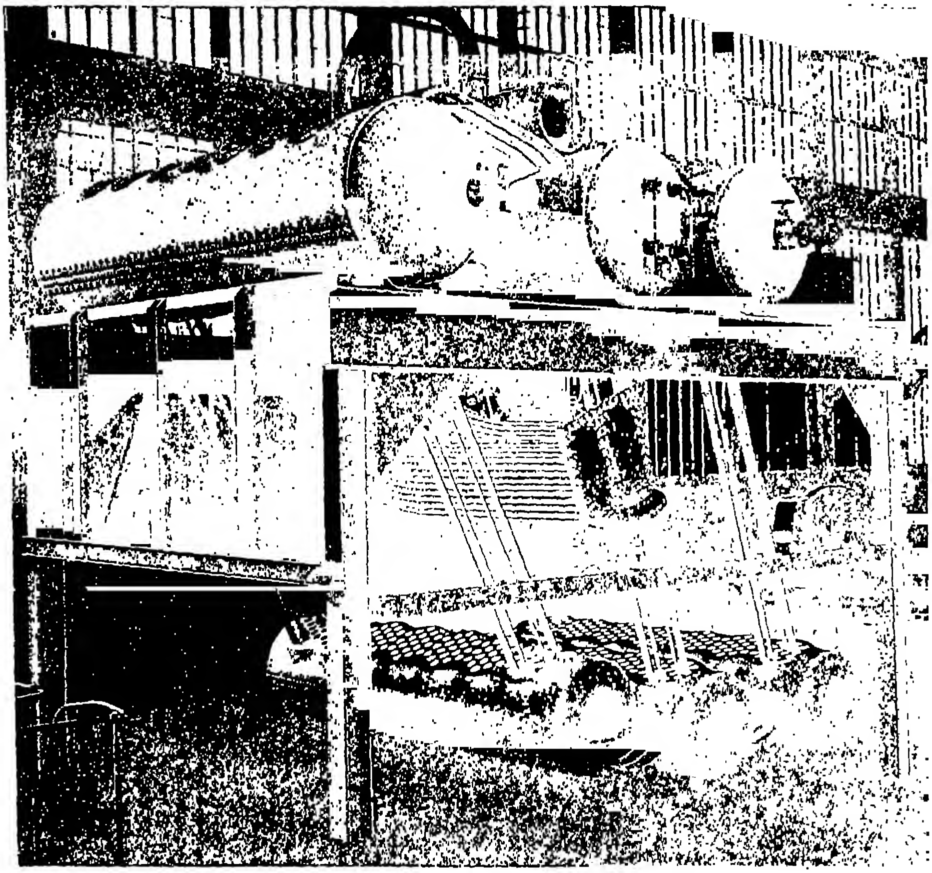


Fig. 38.—Clarke-Chapman Boiler under Construction

length and diameter, the number of spares necessary is a minimum; renewals being made through the manholes in the steam drum. An important point is that the steam generated discharges directly into the steam drum, thus obviating any chance of steam pockets forming. As the tubes are straight the sections must expand as a whole, while the steam drums must be flattened to receive the tubes.

Clarke-Chapman Boiler (Marine Type).—The marine type of boiler is very similar to the land pattern; the main difference is in the head room and absence of brickwork. Fig. 39, which shows the two-section type, is fitted with a superheater. The furnace gases pass up among the front

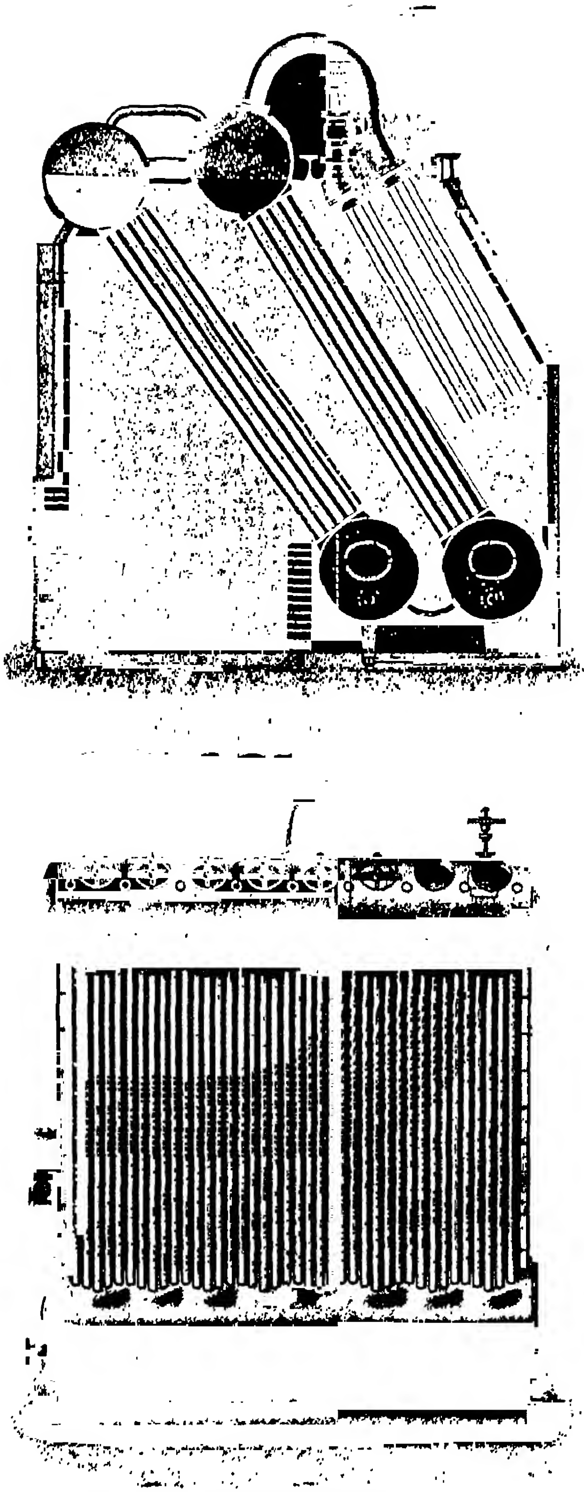


Fig. 39.—Marine-type Water-tube Boiler with Superheater

section of tubes, down the rear section, and up through the superheater tubes to the funnel.

Approximate particulars of a few sizes of the three-section land type Clarke-Chapman boilers are given in the table.

	Overall Sizes.			Shipping Weight.	Heating Surface Area.	Grate Area.
	Height over Steam Drum.	Width.	Front to Back.			
	Ft. In.	Ft. In.	Ft. In.	Tons.	Sq. Ft.	Sq. Ft.
Hand-fired	20 9	6 3	20 3	18	1175	19½
	22 9	9 5	21 0	30	2700	42
	22 9	15 9	21 0	49	5400	83
Chain-grate Stoker	20 9	6 3	24 0	24	1175	21½
	22 9	9 5	26 3	41	2700	52
	22 9	15 9	26 0	69	5400	105

Thompson Water-tube Boiler.—This boiler is in general design similar to the "Woodeson" type. The tubes are all straight and are expanded directly into the steam drum. Each drum has only one manhole at the end, through which all tubes are accessible, thus obviating the necessity of a considerable number of manholes or a multitude of handhole fittings. No extra head room is required for renewing tubes, as these can be passed in and out through the fire-door opening. The ground space is small. These boilers are steady and dry steaming and respond quickly to an overload, and are, therefore, suitable for use in generating stations. There are three standard patterns with one, two, or three pairs of steam-and-water drums, the usual working pressure being from 150 to 200 lb. per square inch.

Stirling Boiler.—The standard Stirling boiler (fig. 40) consists of three steam drums and two water or mud drums connected by four banks of vertically-inclined tubes. The rear steam drum is connected to the middle steam drum by tubes above the water line only, while the two front drums are connected both above and below the water line. The two mud drums are connected in a similar manner. The tubes of solid-drawn steel are slightly curved to permit them to enter the drums radially, and to provide for free expansion. The tubes are arranged in parallel rows, so as to give a clear passage through the boiler. The whole boiler is supported by the steam drums on a steel frame, and is therefore independent of the brickwork. A firebrick arch passes over part of the furnace, the space between it and the front bank of tubes forming a large combustion chamber. This arch being very hot prevents chilling of the boiler by a rush of cold air when the furnace doors are opened. The furnace gases are led by means of firebrick baffles up and down through successive banks of tubes and finally down the rear bank to the chimney flue. The feed water is distributed by means of a feed distribution box throughout the whole length of the rear drum, thus ensuring that all the tubes in the rear bank receive the feed water. The water passes

down the rear bank, depositing any solid matter in the rear mud drum, up the third, down the second, and finally, with the steam formed, up the front bank. The steam is here separated and passed into the centre steam drum through the steam connecting pipe and thence away. The water passes through the connecting tubes below the water line to the centre steam drum and so once again down the second bank of tubes.

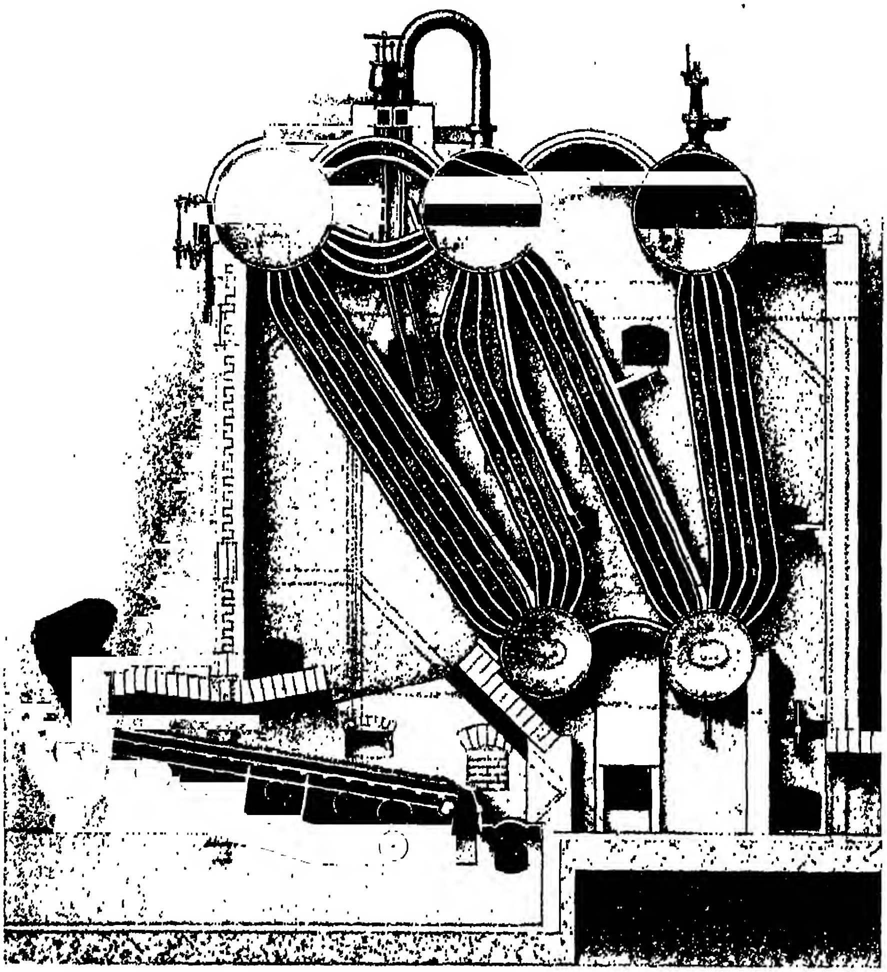


Fig. 40.—Stirling 5-drum Boiler, Superheater, and Stoker

A feature of this design is that the agitation due to steam formation taking place in the front steam drum, while steam is removed from the centre steam drum, facilitates the drying of the steam. The tubes being bent at the ends enter the drums radially, thus obviating any necessity of flattening portions of the drums, while they can expand individually, so preventing stresses due to unequal expansion. The spacing of the tubes

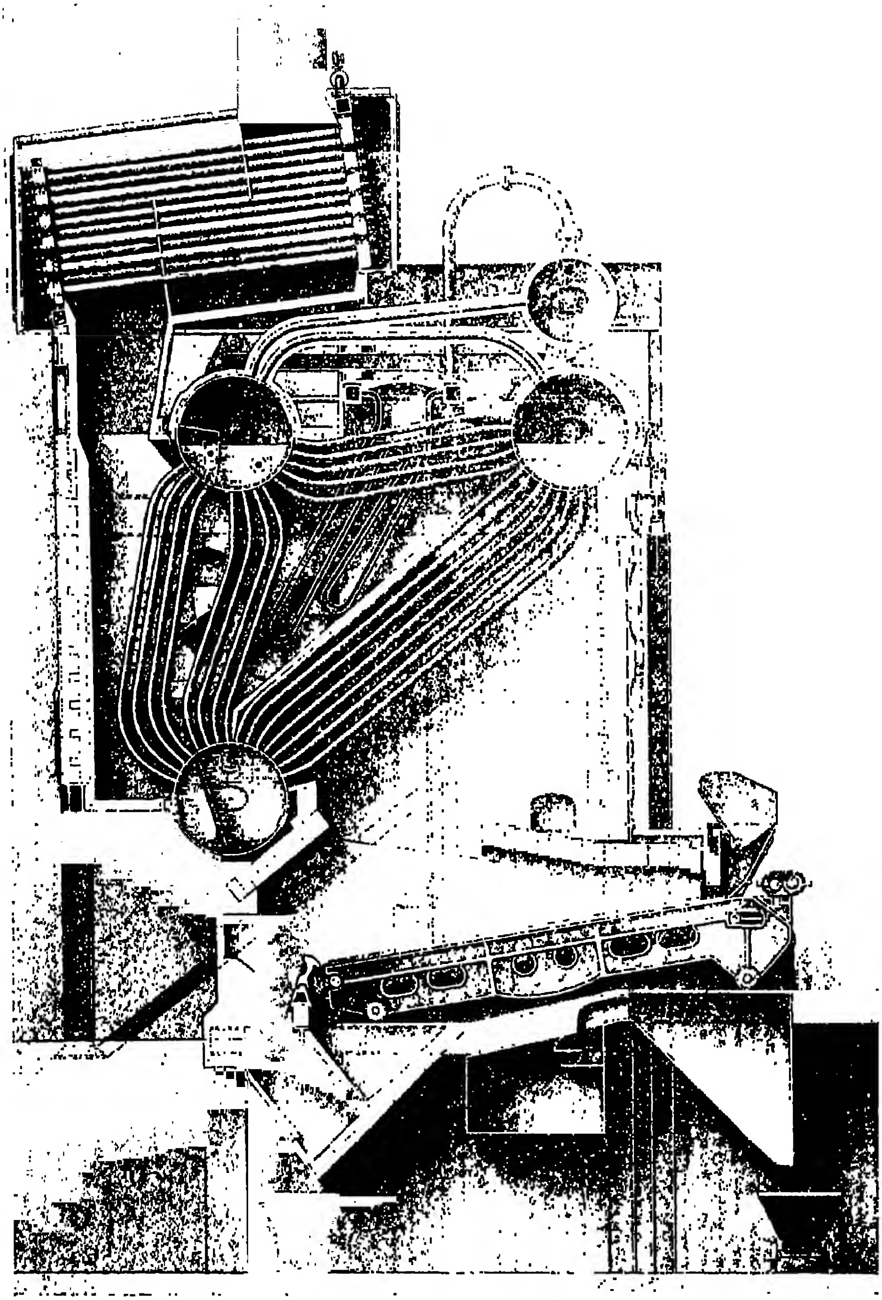


Fig. 41.—Stirling High-duty Tri-drum Boiler, Economizer, and Stoker

is such that any tube can be replaced without removing the others. The fact that the tubes are not perfectly straight complicates inspection and cleaning, and for this purpose special turbine cleaners are supplied by the

makers. The makers claim that owing to the feed water having to pass down the rear bank, which is in contact with the coolest gases, the formation of precipitate is reduced to a minimum, and is mostly confined to the rear bank of tubes, where the danger of failure is very small.

Stirling High-duty Tri-drum Boiler.—This boiler, while retaining the main characteristics of the standard Stirling boiler, contains certain modifications to obtain a high rate of evaporation per square foot of heating surface. It is thus specially suitable for marine work.

Fig. 41 shows the boiler with superheater, economizer, and stoker. It consists of three drums: two upper steam-and-water drums supported on a steel framing, and a lower mud drum suspended from the steam drums by three banks of solid-drawn steel tubes. The two steam drums are connected by a series of horizontally-inclined tubes, both above and below

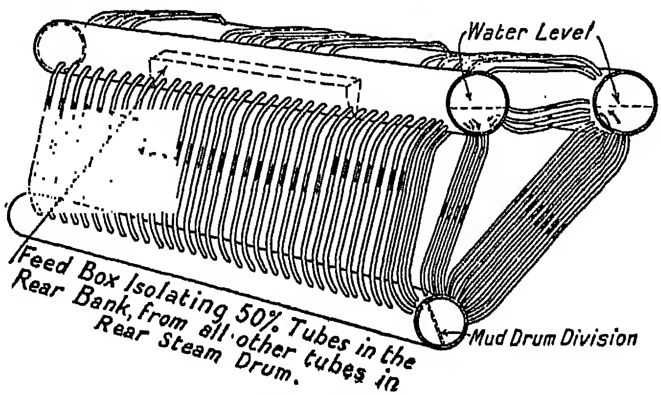


Fig. 42.—Circulation Diagram in High-duty Boiler

the water line. The water circulation is shown in fig. 42. The mud drum contains a division which divides it into two compartments. The rear steam drum contains a feed box, so arranged that the feed water passes from the rear steam drum down half the tubes of the rear bank only. Owing to the mud drum division the water must then pass up the other half of the rear bank of tubes, down the centre bank, and then up the front bank, together with the steam generated, into the front steam drum. The steam is here separated, the water passing back into the rear steam drum through the horizontal tubes below the water line, while the steam passes through the steam connecting pipes and thence into a dry-steam drum. The furnace gases, guided by suitable baffles, pass up the front bank of tubes, down the middle and up the rear bank to the uptake, in which in this case an economizer is shown. In this design nearly half the total heating surface is immediately above the combustion space, and this results in the bulk of the evaporation being performed in the front bank of tubes. The temperature of the gases passing away from the front bank is therefore correspondingly low. The makers state that this type of boiler can be manufactured to evaporate up to 100,000 lb. of water per hour and for pressures up to

500 lb. per square inch, while its weight is about 75 per cent as much as a Scotch boiler of equal capacity. Owing to the shorter path of the furnace gases less draught is required than in the five-drum type.

Yarrow Boiler.—The requirements of a boiler for naval purposes, especially in the case of fast craft such as destroyers and fast cruisers, are small weight compared with the power generated, together with extreme compactness and consequent small space occupied. This has resulted in the production of what is sometimes known as the "Express" type of water-tube boiler, the water tubes being generally of much smaller diameter and larger in number than is the case in those already mentioned. These features are embodied in the Yarrow boiler, which is extensively used in

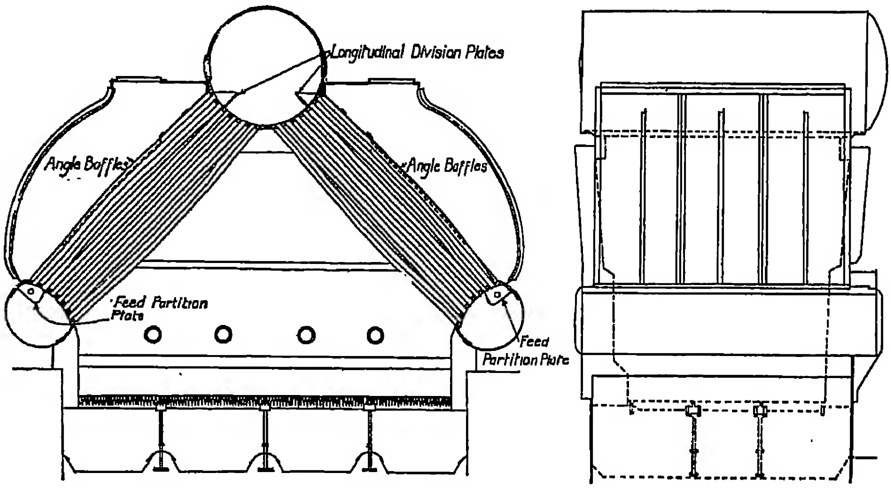


Fig. 43.—Yarrow Boiler

the British and other navies. It consists of a cylindrical steam drum, situated at the top, and two water drums, one on each side at the bottom. The steam drum is connected to each water drum by a large number of straight water tubes. The space between the two banks of water tubes forms a large combustion chamber.

In fig. 43, which illustrates the design of the boilers made for the Chilian battleships in 1913,* a certain number of the outer tubes are divided off by partitions in the water pockets. The feed water entering these compartments passes up the outer tubes, where the temperature is lowest, into the steam drum. By thus compelling the feed water to ascend the tubes farthest from the fire, not only is the economy increased, due to the cold water taking more heat out of the gases than would otherwise be the case, but any impurity in the water is deposited in these tubes rather than in those nearest the fire. In order to ensure that this water is thoroughly mixed, it has been found necessary to fix a longitudinal plate along each side of the steam drum, so as to partition off the feed heating tubes where

* See *Engineering*, Vol. XCV, p. 681.

the feed enters. This prevents the feed water descending the adjoining tubes without first mixing with the contents of the upper drum. The products of combustion pass from the combustion chamber through the two banks of tubes into the uptake on each side of the steam drum, and thence to the funnel. In order to ensure that the gases pass uniformly through the nests of tubes, angle-iron baffles are placed on the outside rows of tubes. They consist of angle irons which lie on the tubes to close up the space between adjacent tubes, and so restrict the area for the flow of gases where they would otherwise pass too readily into the uptake. These baffles are hung on projecting studs and are easily removed for cleaning and inspection purposes. It has been found that the introduction of the baffles has had a considerable effect in securing a high efficiency. The whole boiler is covered by a casing of steel and asbestos.

The results of two trials, reported in the article already mentioned, are given in the accompanying table. The boiler had a heating surface of 4440 sq. ft., with a grate area of 93·9 sq. ft., and the combustion chamber was 610 c. ft.

Nature of Trial.	Duration.	Pounds of Coal per Foot of Grate.	Pounds of Coal per Foot of Heating Surface.	Pounds of Water evaporated per Pound of Fuel.	Pounds of Water evaporated per Foot of Heating Surface.	Temperature of Uptake.
	Hours.					Deg. F.
Official thirty-hours' coal-burning trial	(A) 15	28	0·59	10·96	6.48	622
	(B) 15	17	0·36	13·60	4·93	491

In all cases the evaporation is from and at 212° F.

Recent experience tends to show that feeding the water into the lower drums, while certainly preserving the hotter tubes, accelerates corrosion on the outer tubes where the feed water enters. It may also cause considerable stress in the lower D-shaped water pockets, with the probability of grooving in the pocket tube plate. Yarrow's latest patent water-tube boiler with superheater is shown in fig. 54. It will be noticed that in this case the water drums are cylindrical and that the feed water enters the steam drum immediately above the rows of tubes, which are in contact with the hottest gases, the steam entering the steam drum from these tubes ensuring the thorough mixing of the water. The circulation takes place down the outer and cooler tubes into the cylindrical water drums, and up the inner ones. Loose mud or sediment will collect in the water drums, from which it can be blown out. Zinc slabs are placed in the drums, which by their galvanic action retard any corrosion on the plates and tubes. All exposed parts of the drums are covered with lagging to prevent radiation as far as possible.

Thornycroft Water-tube Boiler.—The Thornycroft boiler has undergone considerable modifications since the design known as the

"Speedy" type was first produced by Sir J. I. Thornycroft in 1885. This boiler consisted of three cylindrical drums arranged similarly to the present Yarrow pattern. The two banks of tubes were, however, curved from the water drums so as to form an arch over the combustion chamber, and then back round to the upper portions of the steam drum, which they entered above the water line. The two groups of tubes were bounded on either side by water-tube walls, formed by two rows of tubes being brought together except where necessary to splay them to allow the passage of the gases. In this way the furnace gases entered the banks of tubes at the bottom, passed along their curved length to the top, and then through the splayed ends outwards to the uptake

The requirement of torpedo-boat destroyers, demanding as large a grate area as possible, led to the introduction of the "Daring" type of boiler,

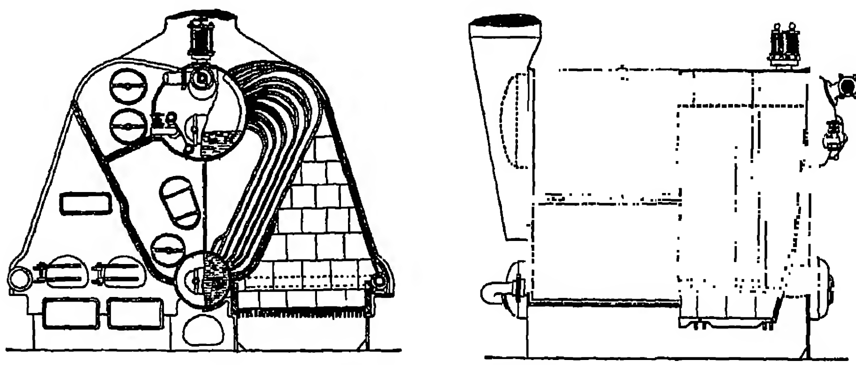


Fig. 44.—"Daring"-type Boiler

fig. 44. In this boiler there are two drums only, one upper and one lower, connected by two groups of water tubes, which enter the upper drums above the steam line. The drums are also connected by a row of vertical downtake tubes of about 4 in. diameter, running down the centre line of the boiler. Two walls of water tubes are also arranged down the outer side of each combustion chamber, the bottom ends of these tubes being connected to a large horizontal water tube which connects to the lower water drum. The sides of the two banks of water tubes also form a water-tube wall to the inner side of each combustion chamber, the furnace gases entering through the splayed-out ends near the lower water drum, then passing diagonally among the tubes into the heart-shaped central space below the upper barrel, and thence to the back of the boiler into the uptake. The steam generated in the water tubes passes direct into the steam space in the upper drum, water flowing down the vertical downtake pipes into the lower drum.

A further modification is the "Tartar" type. In this boiler there are three drums, one upper and two lower, each of the lower drums being connected to the upper barrel by a bank of water tubes of small diameter. The tubes are straight for most of their length, being curved at their ends only. Each

lower water drum is connected to the upper drum by a return tube of large diameter exterior to the furnace chamber.

The latest type of Thornycroft boiler is shown in fig. 45. The upper drum is connected to the two water drums by two banks of tubes of about 1 in. diameter, which are straight for most of their length, but are slightly curved at the ends. The makers consider this essential to allow for unequal expansion of the tubes, while it also allows the water drums to be of small diameter. Besides the reduction in weight, this also reduces the amount

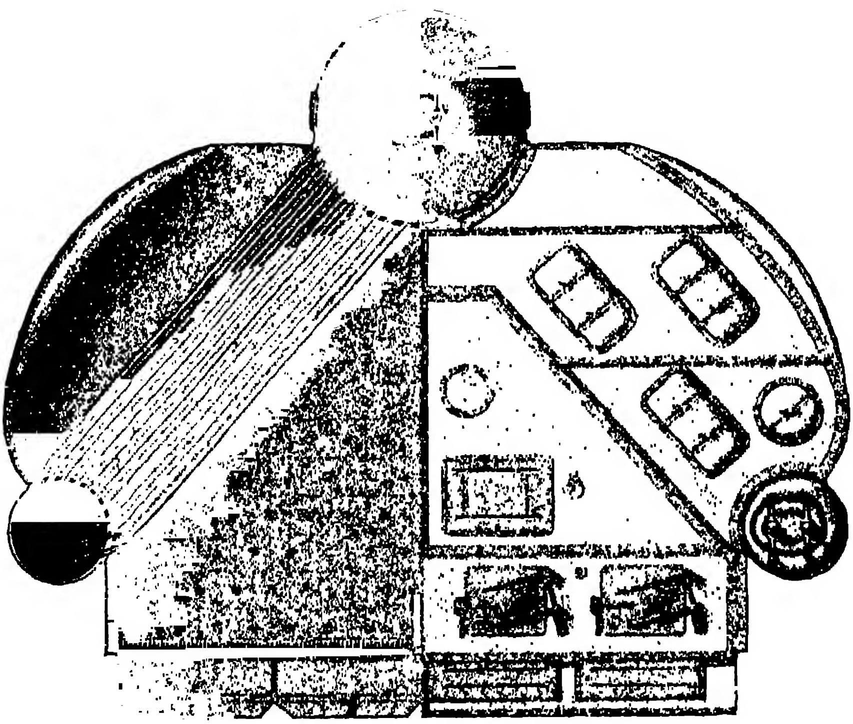


Fig. 45.—Thornycroft Boiler—Coal Fuel

of water in the lower drums to a minimum, thus obviating the risk of dead water and the consequent strains on the drum due to unequal temperatures. The cylindrical form of barrel also reduces the number of joints to a minimum. All three drums are fitted with manholes at the ends for inspection and cleaning purposes. The two lower drums are fitted with blow-down valves. The furnace and combustion chamber are situated between the two banks of tubes, through which the furnace gases pass to the uptake. The two rows of tubes nearest the fire are of somewhat larger diameter than the rest. The combustion chamber is lined throughout with firebrick, the exterior of the boiler being enclosed in a casing of steel plate and asbestos. An air chamber is formed on the two ends of the combustion chamber by means of a steel casing. The water is now fed into the upper drum through a

perforated internal pipe, the perforations being pointed upwards, as it has been found that the practice of feeding into the lower drum and allowing the feed water to pass up the outer rows of water-tubes causes corrosion in these tubes where the water enters.

Fig. 46 shows such a boiler with a heating surface of 1397 sq. ft., and a grate area of 32 sq. ft.

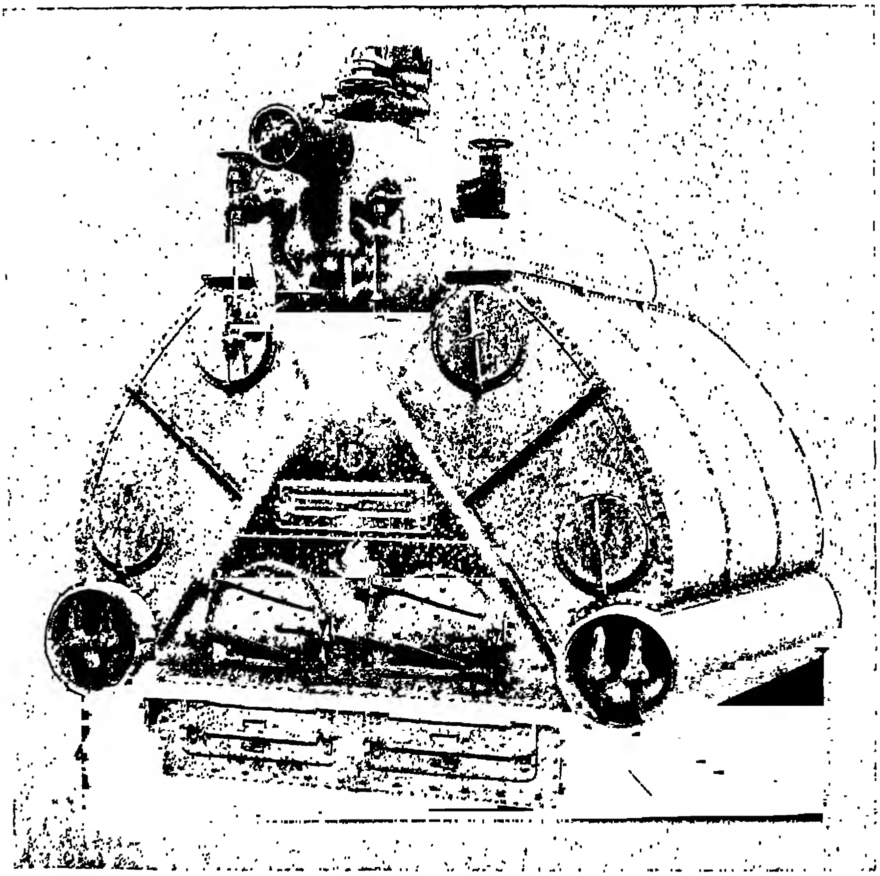


Fig. 46.—Thornycroft Boiler

Mumford Water-tube Boiler.—This boiler, in common with others of the " Express " type, consists (fig. 47) of an upper steam-and-water drum connected to two lower water drums by two banks of water tubes, the fire grate and combustion chamber being between the latter. The steam-and-water drum has in the smaller sizes a bolted cover at one or both ends, and in the larger sizes a manhole. By this means any one water tube may be withdrawn from inside the boiler without disturbing the others. The inner rows of water tubes are curved inwards towards the fire, while the outer ones are curved outwards, a space being left between the two rows of tubes

so formed. An asbestos-covered baffle runs from the steam drum half-way down this space. The course of the gases is shown in the figure. The space between the two banks of oppositely curved tubes is designed to give opportunity for complete combustion to take place before the gases pass to the uptake.

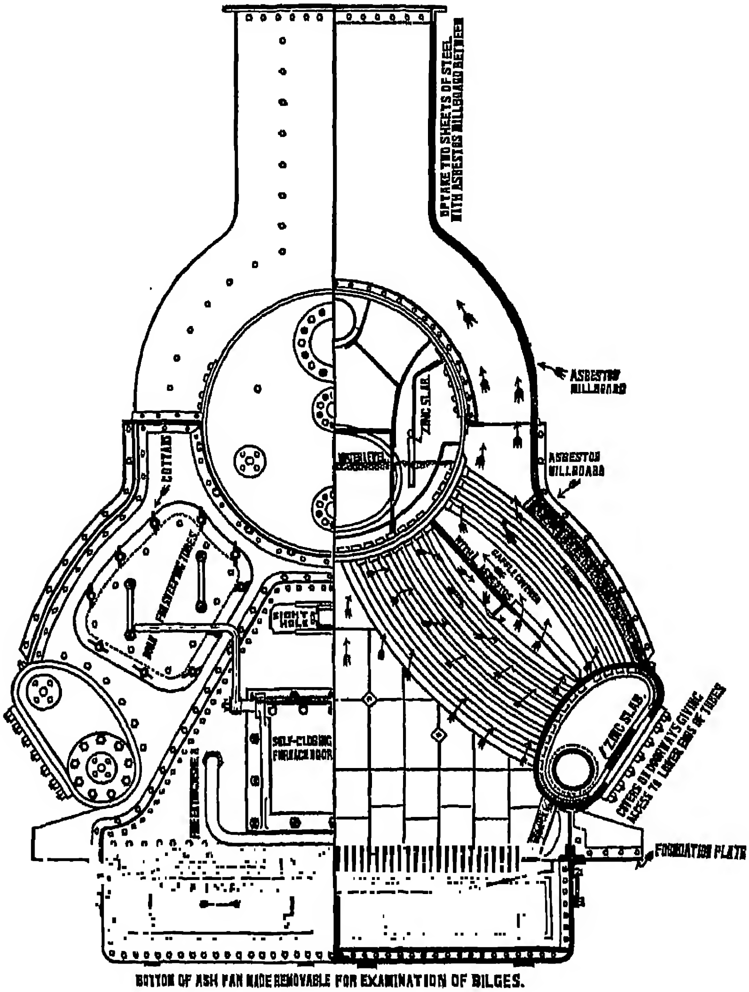


Fig. 47.—Mumford Boiler

The tubes enter the drums radially, the lower water drums being specially shaped to ensure this. The tubes, being curved, allow for expansion, and the curvature in all the rows being the same makes the number of necessary spares a minimum. The outer side of each water chamber has a large opening fitted with bolted covers. It is claimed that any disadvantage, due to the consequent introduction of a flat surface and extra joints, is more than compensated for by the accessibility to the lower ends of the water tubes. The

tubes are not staggered, but are arranged so that there is a clear space between any two rows when looking through them from either direction. This is done to obviate the tendency for soot and small cinders to lodge between the tubes and neutralize the effect of the extra heating surface obtained by staggering the tubes. Cleaning the outside of the tubes is also simplified by this method. The whole boiler is cased with steel sheet and asbestos mill-board.

CHAPTER IV

Superheaters

The advantages of using superheated steam are now generally recognized. Saturated steam, in presence of the water it is generated from, can only exist at one temperature at one particular pressure. If dry saturated steam is heated away from contact with water, its temperature may be raised whilst the pressure remains constant. The efficiency of a steam-engine being dependent, among other things, on the limits of temperature between which it works, greater efficiency can thus be obtained without unduly raising the steam pressure. If heat is abstracted from saturated steam a part will condense, whereas abstraction of heat from superheated steam will first only lower its temperature until it reaches the saturation temperature corresponding to the pressure. The use of superheated steam will therefore save the loss of heat due to condensation in transmission to the engines, and ensure that the steam reaching the cylinders is dry. This is especially valuable in the case of turbines, where water passing over tends to cause erosion of the blading. In the case of reciprocating engines, the condensation due to expansion is also obviated if the steam is sufficiently superheated. The chief difficulties attending the use of superheated steam are the question of lubrication and the effects of the high temperature on the materials of the boiler and engine parts. The use of superheaters on modern land installations is now more or less general. In marine practice, although progress is somewhat slower, there is a steady increase in their application to modern ships.

The design of a superheater depends on the type of boiler with which it is used. Owing to the comparative low heat conductivity of superheated steam, it is essential that the path of the steam in the superheater should be long enough to allow sufficient heat to be taken in by the steam. This point is especially important in the case of smoke-tube superheaters, where, owing to the necessarily small diameter of the superheater tubes, the velocity of the steam will be high. The rate of heat transmission to superheated steam being less than is the case for water under similar conditions of temperature, renders it necessary for precautions to be taken to obviate the risk of the tubes being burnt out. This prevents the superheater being placed in the hottest part of the combustion chamber. At the same time, the temperature

at the chimney end of a well-designed boiler will not be high enough to superheat the steam to any extent. The superheater must also be protected against overheating when steam is not passing through it. This is especially the case when raising steam from cold water. This is sometimes effected by flooding the superheater with boiler water until steam is to be used. An objection to this is that unless the water is pure, incrustation of the superheater tubes may take place. It also necessitates great care that the superheater is drained before being put back into use, to obviate the trouble which would be caused by water entering the main steam line. The design should be such that it can expand and contract freely, in order to obviate any stresses due to changes in temperature. It is necessary that good provision should be made for cleaning the superheater tubes, while it should also be possible to remove part of or all the tubes without the necessity of keeping the

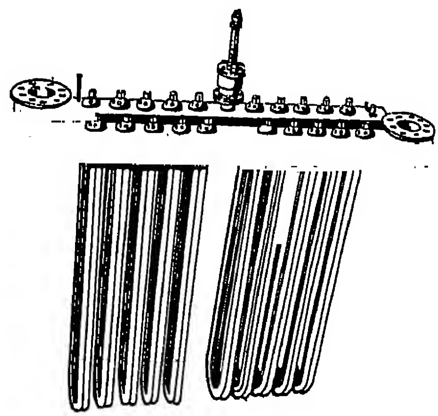


Fig. 48.—Sugden Patent Superheater

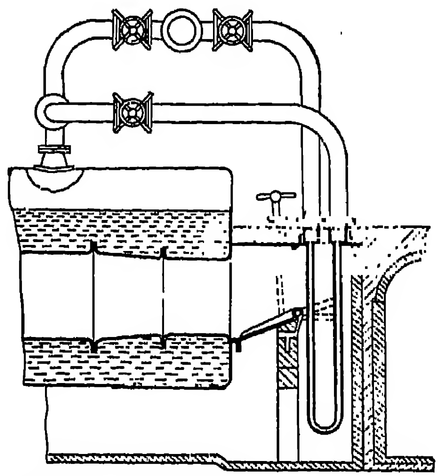


Fig. 49.—Arrangement of Superheater Damper

boiler out of action. In modern boiler installations the superheater is generally a unit of the boiler, but under some circumstances an independently fired superheater is necessary or advisable.

Superheaters for Cylindrical Boilers.—Superheaters for use with boilers of the Lancashire and dryback type are generally of the form shown in fig. 48, which illustrates Sugden's patent superheater. It consists of two mild-steel headers connected by groups of four U-shaped tubes of solid-drawn steel which are expanded direct into the headers. The spaces between the groups of tubes ensure distribution of heat between the front and back portions of the tubes. An oval handhole is fitted in the header opposite the end of each group of four tubes. These handholes are fitted with an external and internal cover, with metal-to-metal joints. Each header is provided with a blow-off cock for draining purposes, and a small test cock for testing. The tubes can be replaced without disturbing the superheater, while emergency tube stoppers can be fitted in the headers in place of a defective tube if required. The arrangement of the superheater in a Cornish or Lancashire

boiler is shown in fig. 49, the main steam-pipe being between the two upper stop valves. With the damper in the position shown in the figure, all the furnace gases pass through the superheater tubes. By raising the damper the gases can be made to bypass the superheater to any desired extent. By raising the damper completely the superheater is entirely shut off from the products of combustion. The arrangement of stop valves allows the steam to be used direct from the boiler if necessary. If the dimensions of the uptake prohibit the use of a damper, Messrs. T. Sugden use a special circulating device. In this case, while raising steam the outlet side of the superheater is connected to the boiler by means of a small pipe entering the boiler above the water line through a non-return valve. As steam is generated in the boiler it will flow through the inlet pipe to the superheater, the superheated steam returning to the boiler through the small pipe. When steam is being used the non-return valve closes automatically. In the dryback type

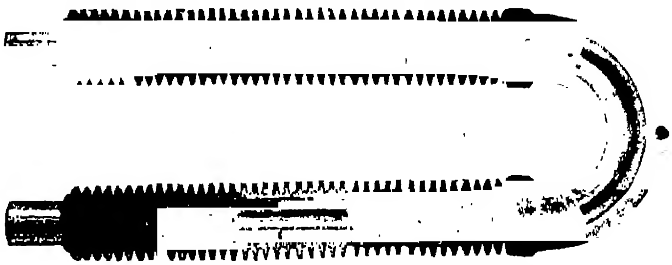


Fig. 50.—Part Section of Foster Superheater Element

of boiler the temperature immediately behind the boiler is generally too high for the superheater. The best arrangement is then to make the superheater in two distinct sections connected by cross-over pipes, a section being placed against either wall of the flue.

Foster Armoured Superheater.—This superheater is especially designed to withstand a large amount of heating, with a view to avoiding the necessity of flooding devices during steam raising and stand-by periods. It consists of a series of U-shaped elements parallel to each other, the ends of which are directly expanded into wrought-steel headers arranged in the usual way. The elements (fig. 50) consist of cold-drawn steel tubes, over which are expanded a series of cast-iron gills with metal-to-metal contact. The mass of metal covering the tubes will serve as a reservoir of heat, thus imparting a more or less even temperature to the steam, in spite of fluctuations in the temperature of the hot gases. Core pipes, supported centrally by small knobs, are fitted inside the elements, forming a thin annular passage for the steam, which is thus forced to travel in a thin layer close to the heating surface.

"North-Eastern" Superheater.—This superheater, which resembles the Schmidt type, is especially designed for the Scotch marine boiler. It consists of headers, fitted in the smoke-box, arranged in pairs between the nests of boiler tubes as shown in fig. 51. One of each pair of headers is a

saturated steam header and is connected by pipes to the boiler stop valve; the other, being the superheated steam header, delivers the superheated steam to the main steam-pipe supplying the engines. The superheater tubes or "elements" connect the saturated and superheated headers, and consist of a series of small tubes of U shape carried in and out of the boiler smoke tubes. In order that the steam should have ample opportunity to

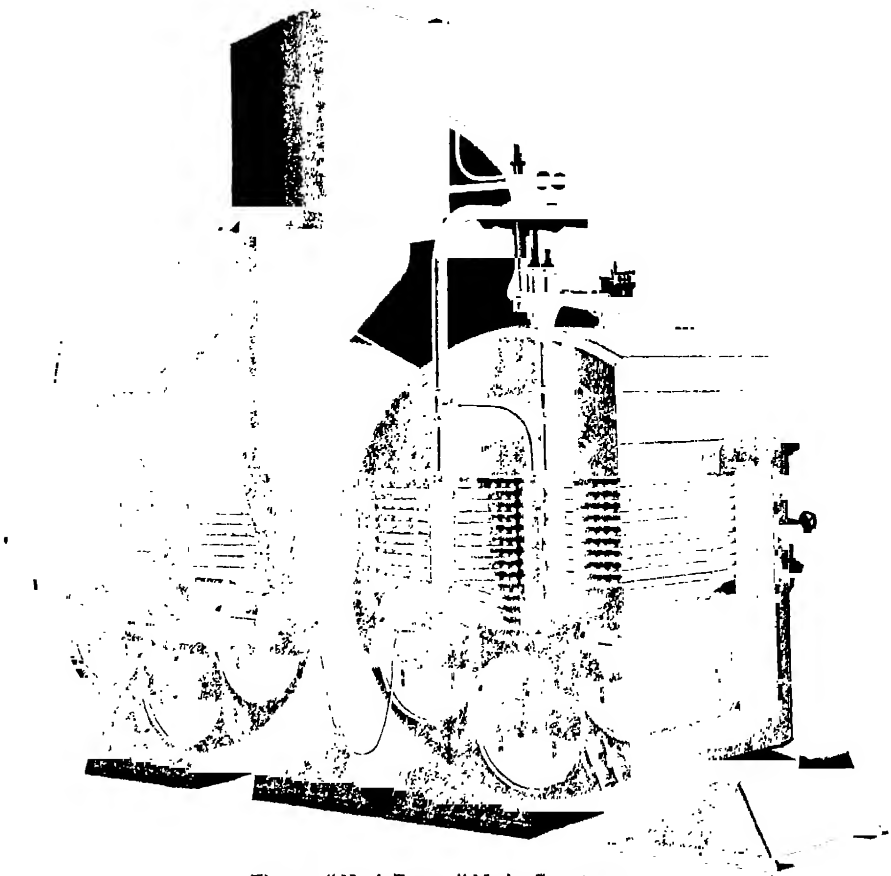


Fig. 51.—"North-Eastern" Marine Superheater

take in heat, there are generally about four of these tubes connected in series between each pair of headers, as shown in fig. 52. As the ends of these tubes which project into the boiler tubes are subjected to the hottest portions of the products of combustion, the U bend in each pair of tubes consists of a special steel cap of suitable form welded to the tubes. The connection between the element and the header is made by fixing a collar to the end of the tube, this collar fitting into a recess in the header and being provided with a jointing ring. A steel clamp, screwed down to the header by means of a stud and nut, holds each pair of collars in place and makes a steam-tight joint. All the headers are provided with drain valves for blowing out accumu-

lated water while raising steam. By means of stop valves the superheater can be isolated, if necessary, to remove any superheater element, holes left by the removal of the elements being blanked off by steel plugs. A pyrometer can be fitted to the superheater as a check on the firing of the boiler.

A design of superheater situated in the uptake of the Scotch marine boiler is made by Messrs. T. Sugden, which will provide from 50° to 100° of superheat. It consists of two headers situated above the boiler and connected by a series of superheater tubes bent backwards and forwards, and passing down the uptake as far as possible. As there is room enough

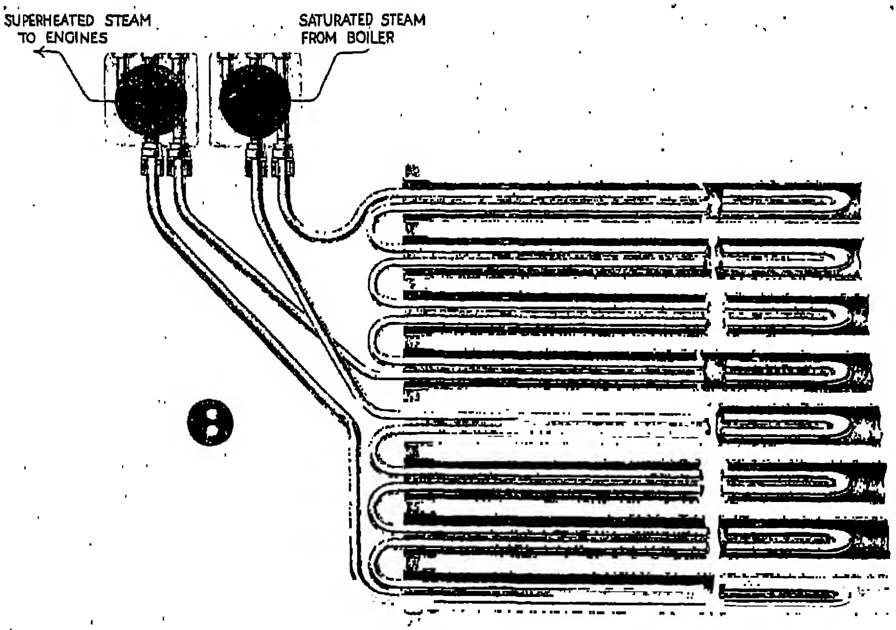


Fig. 52.—Details of Superheater Element

in this to provide a considerable heating surface in the tubes, a fair degree of superheat can be obtained by using the heat from the waste gases of the uptake. The headers being out of the path of the gases, the joints are not liable to damage by heat. This type of superheater can be fitted with slight alteration to the existing plant, and is especially useful with natural draught boilers if only a moderate amount of superheat is required.

Locomotive Superheater.—There are four positions on the locomotive in which a superheater might be placed, namely, in the fire-box, boiler, smoke-box, and the smoke tubes. No practical arrangement of the superheater in the fire-box has been devised, owing to the difficulty in arranging sufficient protection from the great heat, while the placing of the superheater in the boiler is undesirable owing to the extreme difficulty of getting to it for repairs and cleaning. Superheaters placed entirely in the smoke-box have the advantage that the otherwise wasted heat from the furnace gases

is used, but in a well-designed boiler the temperature of these gases is not sufficiently high to effect a high degree of superheat. For this reason, in some designs the superheater placed in the smoke-box is heated by part of the products of combustion passing through a large flue which is situated in the lower part of the boiler. Modern locomotive superheaters are mostly designed with the superheater tubes situated in the smoke tubes, and extending into headers situated in the smoke-box.

Of such a type is the Schmidt superheater, fig. 53, to which most modern designs approximate in general features, although details differ to a consider-

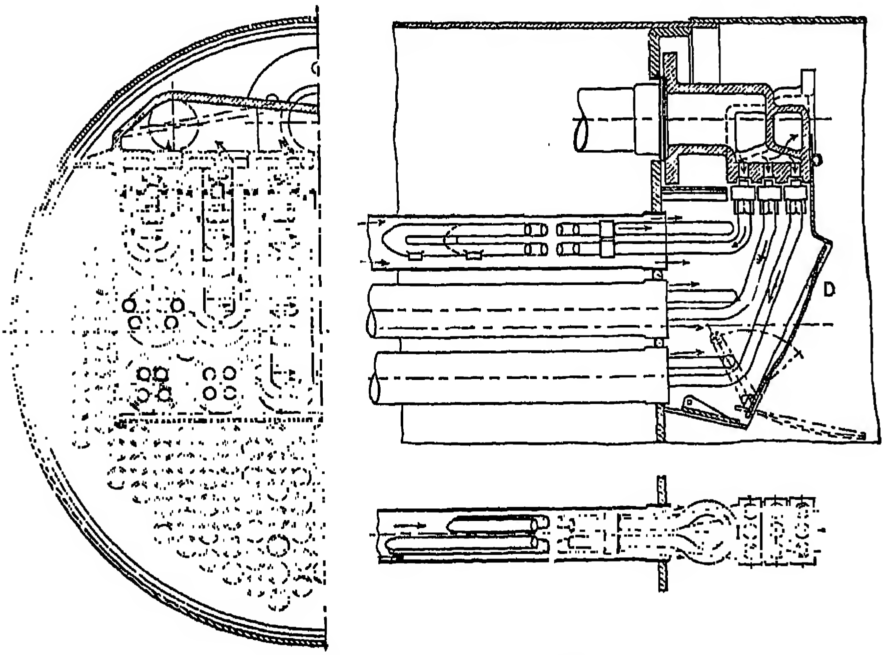


Fig. 53.—Schmidt Locomotive Superheater

able extent. The three upper rows of fire tubes are of larger diameter than usual. Each large fire tube contains a superheater element consisting of a weldless steel tube arranged in a double loop, so that the steam passing through this tube traverses the portion of the fire tube four times. The superheater elements are connected to a chambered header, so that saturated steam from the boiler passes through alternate chambers to each superheater element, and after being superheated by passage through the element, passes into the other alternate chambers of the header, and so to the cylinder. The elements are generally connected to the headers by bolts, and can be removed separately, the holes in the headers being blanked off by inserting a flanged bolt. The ends of the superheater elements nearest the fire are either of special thick section, or the end bend is made by a separate end piece welded on to the straight tubes. Originally, in order that the tubes should not be burnt when steam was not being used, a damper D was arranged in the

smoke-box. This damper could be closed when the engine was stationary, and so prevented the furnace gases passing along the smoke tubes containing the superheater elements. It has been found, however, that a small amount of steam passing through the superheater tubes gives sufficient protection against burning out, and the damper is now dispensed with. In the Robinson superheater the necessary steam is circulated through the elements by means

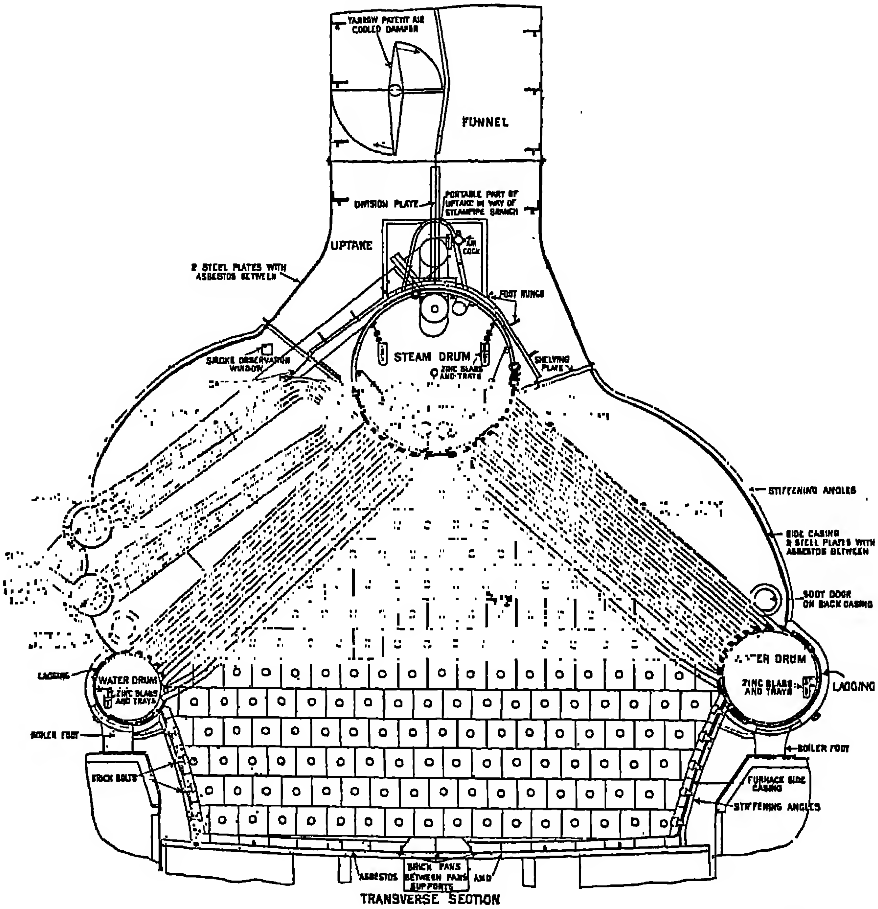
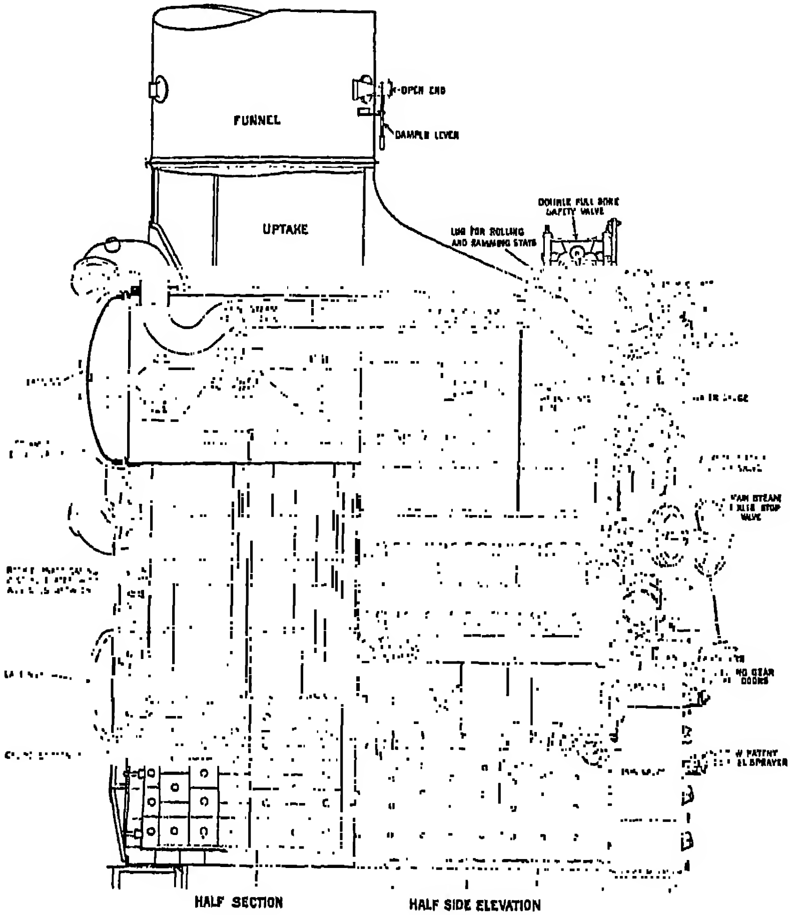


Fig. 54.—Yarrow

of a circulating valve which provides for a small amount of steam to pass from the boiler to the distributing side of the header, through the elements, and finally to discharge through the blast pipe. In addition to this the return bends of the elements are solid forgings through which the steam passages are machined, experience showing that this type of end will withstand the hot gases passing through the fire tubes. In its general arrangement the Robinson superheater resembles the Schmidt type, the chief difference being the design of the headers into which the superheater tubes are expanded. The Horwich superheater has separate headers, the saturated steam header

being arranged as a casing on either side of the main steam-pipe, while the superheated headers are directly over the steam chests of the cylinders.

Yarrow Water-tube Boiler with Superheater.—Mr. H. E. Yarrow has published his views on the application of superheaters to the Yarrow type of boiler in a paper he read before the Institute of Naval Architects, 28th March, 1912. An interesting set of experiments will be found there on the



Boiler

distribution of temperature through the tubes on a standard Yarrow boiler.

In the original paper the superheater was placed on one side of the boiler, as shown in fig. 54, and consisted of two drums connected by a number of U-shaped tubes. In the more recent design the two drums are replaced by a single drum-shaped header separated into compartments by diaphragms. These boilers find their most extensive application in the navy, especially in destroyers, where extreme lightness has to be associated with very large powers. This consideration will be realized when it is remembered that the horse-power developed on a modern destroyer is something of the order of

30,000 h.p., and the *total* weight of the machinery in the whole ship is only some 600 tons. The question of the use of superheat in the navy is one which has exercised the attention of responsible officials for many years, but it is only in recent years that any definite tendency has shown itself in the direction of its use. One of the prime difficulties that the engineer of a destroyer has to face, when steaming at high speed with very high rates of evaporation in the boilers, is that of priming, with consequent entry of water into the turbines, and possible destruction of the turbine-blading. It has been very fully realized that use of the superheater will tend to reduce these difficulties, and on this score it would be an advantage, but the advantage would be secured at some additional weight of the boiler plant. The additional weight is difficult to estimate, but where every pound of machinery is studied critically in view of keeping the total weight within the prescribed limits, it has not been considered, at any rate till comparatively recently, that the additional weight is justified.

Babcock & Wilcox Independently-fired Superheater.—The heating surface of this superheater (fig. 55) consists of a large number of vertical seamless tubes of $1\frac{1}{2}$ in. outside diameter and large thickness. The steam entering the pipe A is distributed to a number of horizontal manifolds, which are interconnected by a series of tubes G, in such a manner that a thorough circulation of the steam is obtained before passing off to the collecting pipe H. The furnace is an ordinary grate in an enclosed chamber. The furnace gases pass through a perforated wall L into a muffling chamber O, where the temperature is equalized. This muffling and the consequent even temperature obtained has the advantage of greatly increasing the life of the tubes. The gases then pass through the superheater tubes round the brick baffles M, through the damper into the main flue. There are no screwed connections, the tubes being expanded and bell-mouthed into holes in the wrought-steel connectors. Opposite each tube a handhole is fitted. All tubes being of the same dimensions a minimum of spares is necessary. Particulars of a few sizes are given below.

Dimensions (feet), including Brickwork.			Weight of Superheater (packed).	Heating Surface.	Steam to be Superheated (pounds per hour). Superheat.		
Length.	Width.	Height.	Cwt.	Sq. Ft.	120° F.	200° F.	300° F.
$11\frac{3}{4}$	$4\frac{1}{2}$	$12\frac{1}{2}$	95	198	5100	3500	2500
$16\frac{1}{2}$	$7\frac{3}{4}$	$16\frac{1}{2}$	270	1140	29200	19800	14200
$22\frac{1}{2}$	$14\frac{3}{4}$	16	710	4140	—	—	51500

Babcock & Wilcox Superheater (Integral with Boiler).—This superheater is shown in fig. 56. It consists of a number of tubes bent into U shape and expanded at both ends into manifolds, one of which receives the saturated steam from the boiler, the other collecting the steam

after it has traversed the superheater tubes and delivering it to the steam main. The furnace gases must first pass through the front part of the boiler tubes, which comprise a considerable heating surface, and the superheater tubes are therefore not subjected to the gases at their highest temperature, and there is also no danger of detrimental condensation of the

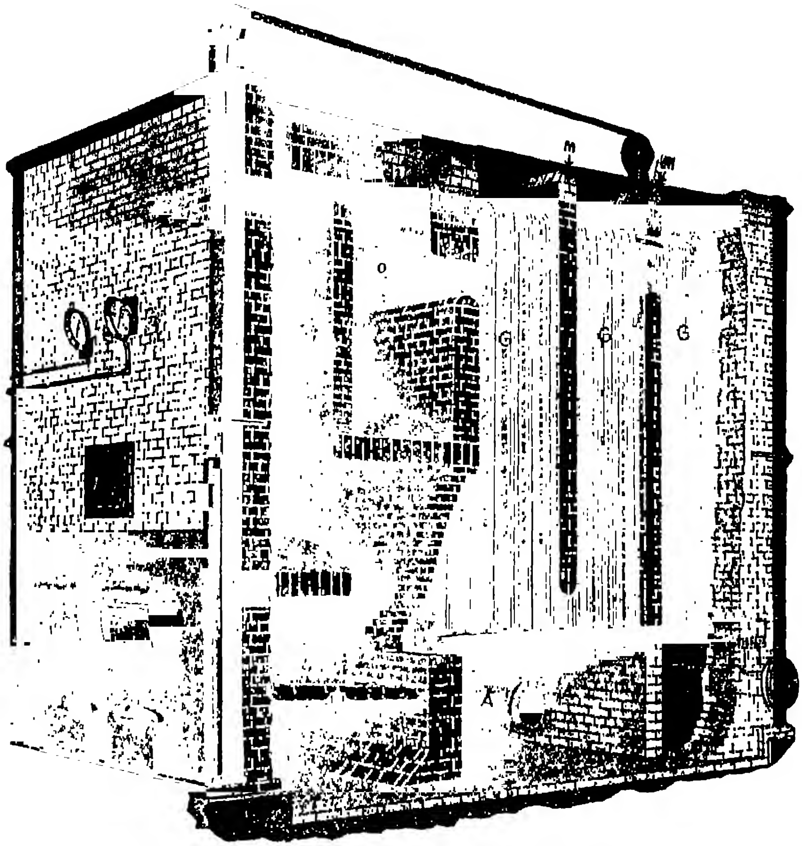


Fig. 55.—Babcock & Wilcox Independently-fired Superheater

A, Main connecting pipe. G, Tubes. O, Muffling chamber. M, Brick flame baffles.

gases. If the boiler is working regularly no fluctuations of temperature will occur where the superheater is situated, and a temperature of 100° to 250° F. of superheat can be obtained. Freedom for expansion is obtained by the tubes being free at one end and by the manifolds not being rigidly connected to each other. To provide against overheating when raising steam from cold water, the superheater can be flooded with boiler water. The flooding arrangement consists of a two-way cock, by means of which water from the boiler can be allowed to enter the superheater tubes through the lower manifold. Any steam forming in the superheater tubes passes into the boiler through the pipes which normally

convey saturated steam to the superheater through the upper manifold. When steam is raised and the superheater is to be used, the water is drained from the manifold through the flooding pipe and the two-way cock.

In the marine boiler* the superheater is placed above the boiler, a vertical baffle being arranged in the superheater chamber so that the gases passing up the rear superheater tubes descend through the front tubes

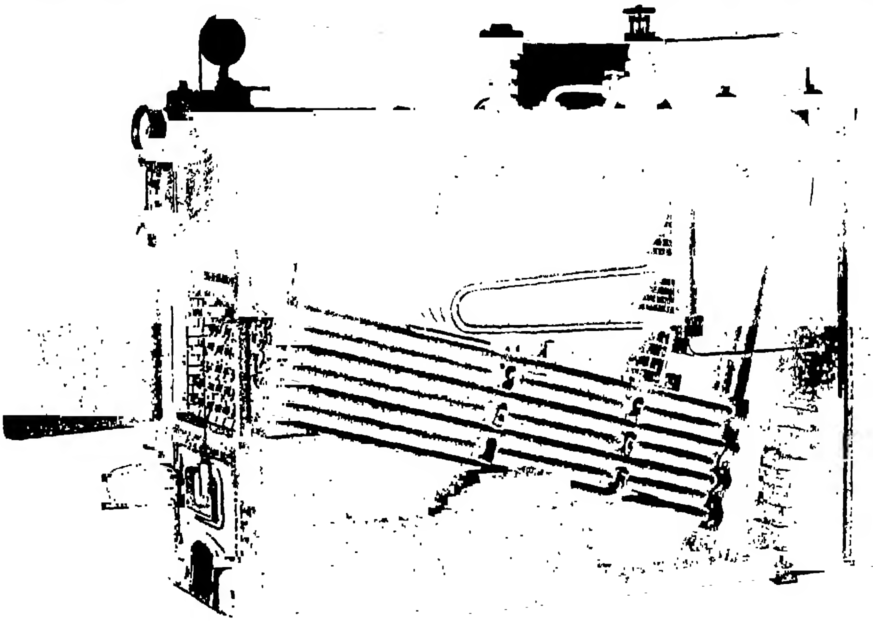


Fig. 56.—Superheater Integral with Boiler

into the boiler proper and proceed as usual. This baffle can be raised, and by so doing a portion of the gases will bypass the superheater. In this way the proportion of the gases passing through the superheater tubes can be adjusted to suit the rate of working of the boiler.

CHAPTER V

Mechanical Stokers

Modern methods of mechanically firing fuel to boilers, while perhaps not completely capable of such instantaneous adjustment to changing conditions as firing by hand provides, possess considerable advantages. With poor types of fuel the mechanical stoker is undoubtedly superior, in

* See paper by A. Spyer, M.I.C.E., &c., read before Liverpool Society of Engineers, March, 1920.

fact, one may say almost a necessity. The ideal aimed at in good stoking is to obtain uniform heat over the fire with complete combustion of the fuel, to eliminate, as far as possible, the production of smoke. At the same time, it should be possible to adjust the firing when necessary, to allow for sudden increase in load on the boiler. The question of smoke prevention has not yet been satisfactorily solved, and the claims of mechanical stokers in this direction are open to doubt. The problem of efficient combustion is one of providing the correct amount of air for the particular type of fuel used. Although a mechanical stoker can be adjusted to give a continuous feed of fuel under constant conditions, yet this very mechanical regularity is a disadvantage where accommodation to different classes of fuel or varying loads on the boiler are called for. For this reason alone it is often held that really efficient hand firing is superior to any mechanical contrivance. In a mechanical stoker the necessity for opening the furnace doors, with the consequent inrush of cold air, is obviated, while a good design gives freedom from clinkering, and the continuous movement of the fuel ensures a clean grate. While they provide a saving in labour and ensure a regular feed under consistent conditions, yet efficient supervision is necessary if economy is to be preserved.

Although there are a considerable number of efficient designs of mechanical stokers, they conform generally to three broad types, namely, coking, sprinkling, and under-feed stokers. In the coking method of stoking the fresh coal is placed on the front end of the fire near the dead plate. Here the volatile products are distilled off, and passing over the incandescent part of the fire are completely burnt. The coal, thus reduced to coke, is pushed along the fire, and by the time it reaches the rear end of the furnace combustion is complete. In the sprinkler method the coal is spread in a more or less uniform manner over the whole surface of the fire. With the underfeed stoker the fresh coal is fed into the furnace from below.

Chain-grate Stoker.—This type of stoker, which is of the coker pattern, is shown in figs. 40, 41, and 77. Owing to its construction it can only be used for externally fired boilers. It consists of a broad endless belt made up of short cast-iron links passing over two drums or sprocket wheels, one outside and one inside the furnace. The drum outside the furnace is driven by mechanical means through a ratchet wheel or worm and worm wheel, causing the top side of the chain belt to move slowly inwards. The speed of the chain can be adjusted to suit the type of coal and conditions of combustion required. The coal is fed into a hopper, through which it descends on to the moving chain, and so passes into the furnace, the thickness of the layer being adjusted by an adjustable trimming plate. The fuel is coked as it enters the furnace and eventually becomes incandescent, the ashes dropping over the front into the ashpit. This type of stoker is simple in construction, and can be adjusted to deal with inferior qualities of coal. The whole machine is mounted on wheels so that it can be removed bodily from the furnace.

The Babcock & Wilcox Chain-grate Stoker.—The special

features of the latest designs of this standard stoker embody a cast-iron grate link of robust construction, particularly designed to prevent the admission of air where the links pass over the drum, and so arranged that the whole belt forms a continuous covering round this drum without exposing an open joint where it passes round the curve. At the same time, on passing round these drums a scissor action between the links is introduced, which effectually clears them of any small coal or ash which may have accumulated between the links during the progress of the grate through the furnace. The upper portion of the grate was formerly carried on rollers, but it is now standard practice to substitute angle-iron runners for these rollers, as by this means the amount of riddling taking place is reduced to a minimum, the grate travelling steadily and smoothly through the furnace the whole time, instead of the links slightly rising or falling as is the case when they are supported by transverse rollers only. This undulation was accountable for a certain amount of riddling before the angle-iron runners were introduced. No additional power is required due to the substitution of these flat angle-iron runners for the rollers. The under part of the chain is carried on rollers as hitherto. The rear end of the chain is covered by dumping bars of special design to ensure that sufficient obstruction shall be offered to the fuel bed in order to prevent it being carried into the ashpit too freely, but at the same time to avoid the accumulation of unnecessary clinker. To meet special conditions of fuel, the angle of resistance offered by these dumping bars can be varied, but a standard bar has been developed to meet all general conditions of running (see fig. 2, OPERATION OF LAND POWER PLANTS, Vol. V).

The driving gear for these stokers is actuated by a gear box in which the teeth are machine-cut and of similar construction to the change-speed gear of a motor-car. Four speeds are arranged for, approximately equal to a grate travel of 6, 12, 18, and 24 ft. per hour. The depth of the fuel bed is regulated by a specially-constructed guillotine door, bevelled at the bottom corners to allow of a slight thickening of the fuel bed at the sides of the grate.

The standard stoker is constructed for natural draught conditions, but is equally applicable for forced draught, as where this is required very slight modifications are made to the side frame of the stoker, so that closing-in plates can be fitted at the front end and the whole stoker boxed in. Air pressure is then introduced into the ashpit, and by this means it is possible to consume very low-grade fuels at a high rate of combustion per square foot of grate area. At the same time, when the forced draught is not required, it is only necessary to open the two front ashpit doors to re-convert the stoker to ordinary natural draught conditions.

Where, for any particular reason, it is not considered desirable totally to enclose the stoker, a specially modified design has been introduced which provides for an air chamber under pressure between the upper and lower chain. This air chamber can be subdivided, and by this means a definite pressure introduced over one or more sections of the grate.

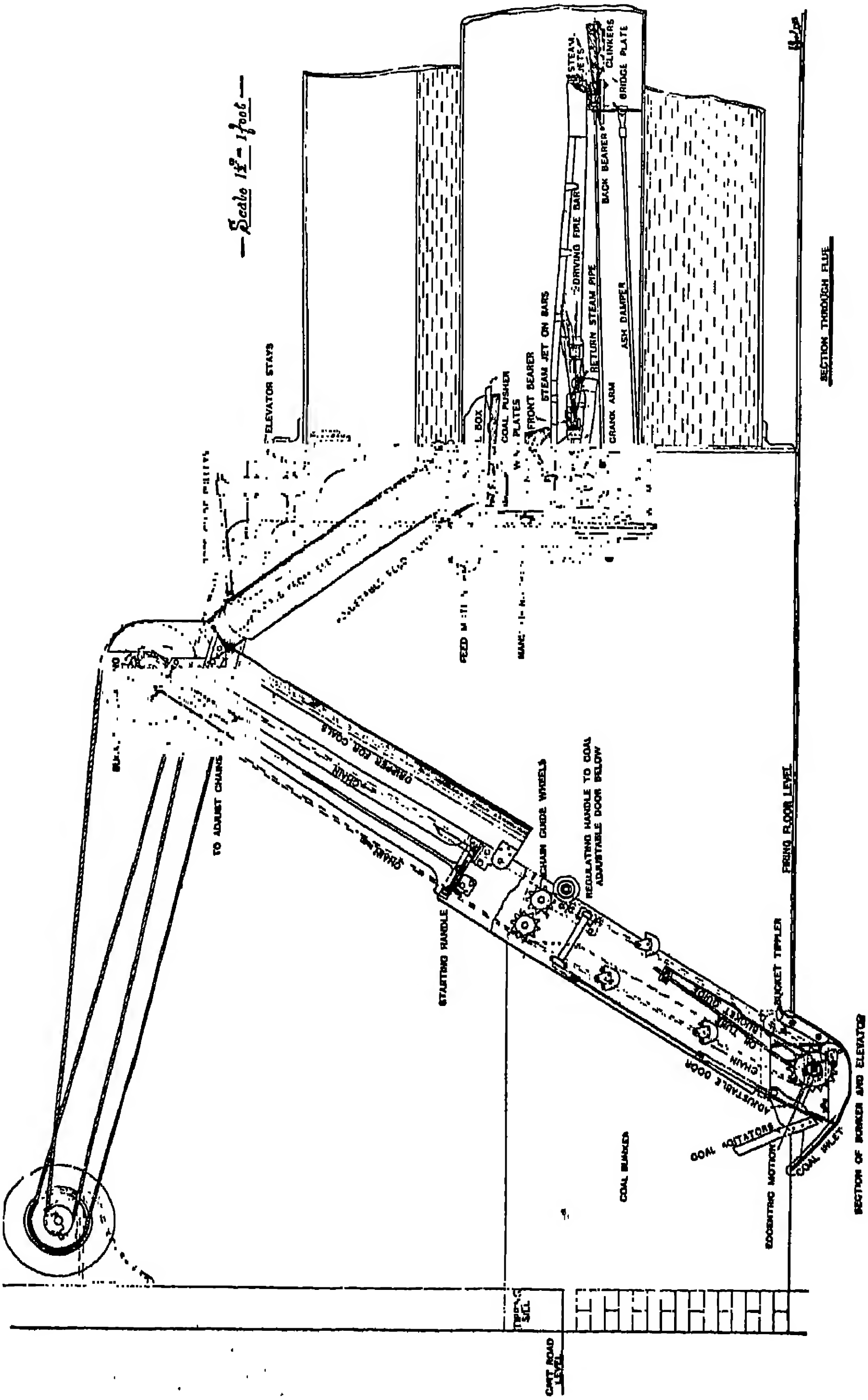


Fig. 57.—" Auto " Stoker and Elevator

"Auto" Stoker and Elevator.—This stoker (fig. 57), made by Messrs. John Ruscoe & Co., is of the coker type.

Motion is transmitted by means of rope gearing to a three-grooved pulley situated on the front of the boiler. This pulley, by means of a crank, actuates a shaft along the hopper front. The coal from the hopper is pushed forward by a coal pusher actuated from the shaft, which can be regulated to any desired rate of feed. The coal falls from the fuel box on to the fire-bars, which are placed parallel along the furnace and are ramped at intervals to form slightly inclined steps. By means of a crank shaft supported along the front of the boiler and actuated through a worm gear from the pulley, four of the firebars have a longitudinal oscillating movement imparted to them, the remainder being driven by knuckle-jointed bar keys. The various motions are arranged so that the side bars have a greater motion than the centre. In this way the bars automatically clear themselves even if the rate of combustion is comparatively high. The fuel is gradually pushed forward, due to the movement and the inclined bars, the residue being pushed over the bridge end of the furnace. This type of stoker, which is of comparatively simple construction, is especially adapted for use in the Lancashire and Cornish types of boiler. It allows for the provision of a fire door, so that hand firing can be resorted to if necessary. The coal can be automatically fed to the hopper by an elevator consisting of an endless chain carrying buckets.

Triumph Mechanical Stoker.—This stoker (fig. 58) is of the sprinkler type. Power is supplied from a suitable countershaft to two pulleys S and SS. These pulleys transmit a slow rotary motion independently to two shafts X and Y by means of worm gearing. The shafts X and Y respectively actuate the feed and the firebar motion, and can be adjusted independently. The coal descends from a hopper A into a feed box in which an adjustable coal pusher C is situated. This coal pusher receives a reciprocating motion from the shaft X through a suitable train of spur wheels. At regular intervals the pusher feeds a small charge of coal to the front of the shovel D, which is held by forged steel arms keyed on to a shaft working in bearings on either side of the feed box. On the centre of this shaft is keyed a steel trigger G. The shaft X carries a four-armed cam F, the projections of which severally engage with the trigger G on the shovel axle, thus raising the shovel arm into the backward position. A prolongation on the shovel arm is connected to a powerful spring E, which is compressed when the shovel is in the backward position. As the cam F rotates and releases the trigger on the shovel arm, the spring drives the shovel smartly forward, scattering the charge of coal over the fire. The four arms of the cam, being of different lengths, vary the throw of the shovel, thus distributing the fuel uniformly over the grate. The firebars are actuated by cams J, threaded loose on the square shaft V. A separate cam for each firebar engages in a renewable nose piece on the bar and draws each bar forward two inches separately and one after the other. This separates it from clinkers and frees the air space, but does not drag forward any portion of the fire. When all the

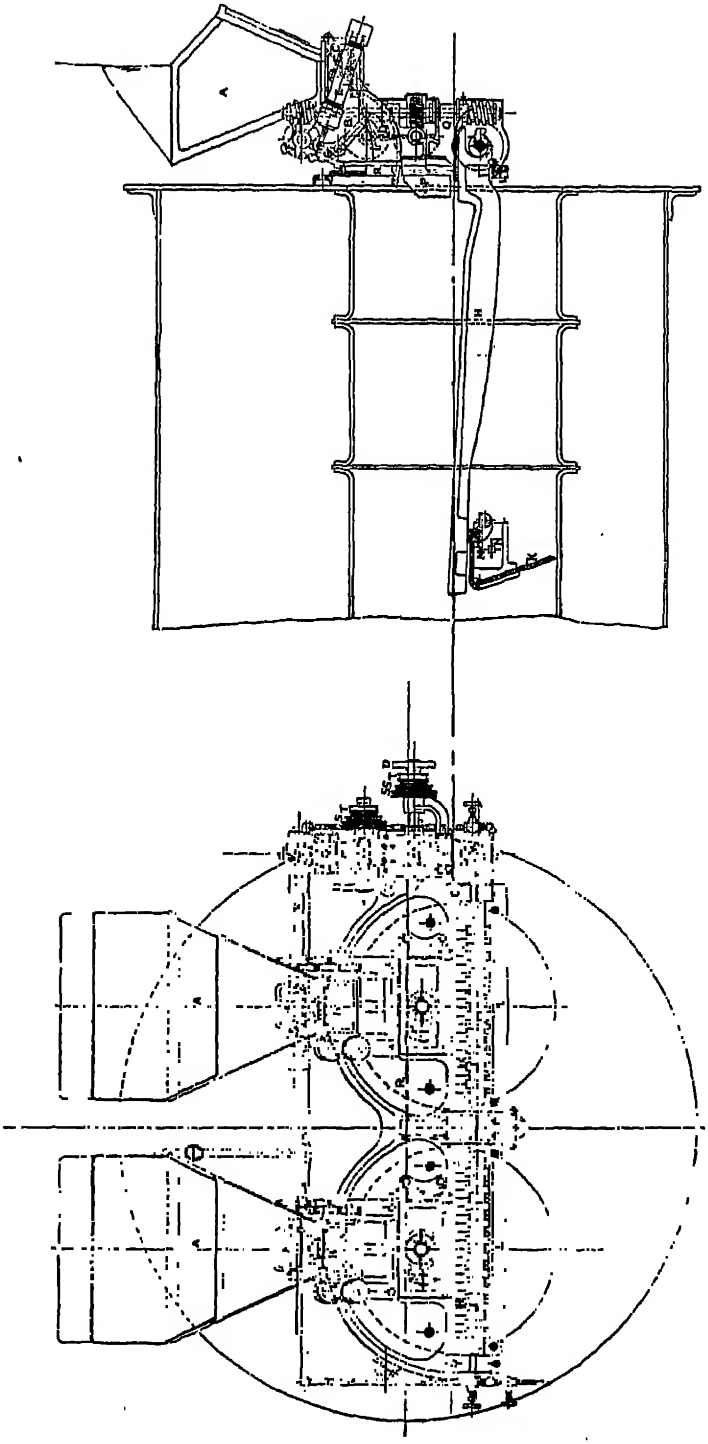


Fig. 58.—"Triumph" Mechanical Stoker

bars have been brought forward, there is a slight pause, and two side projections on the cams then push all the bars simultaneously back into their original position, carrying back with them the whole fire. In this way, the clinker and ash travel towards the back of the furnace and finally fall into the ashpit.

The firebars consist of a cast-iron trough (fig. 59) made the full length of the grate, with bell-mouthed openings at the front end into which superheated nozzles for superheated steam jets are fitted. The trough is fitted with four renewable grids having graduated air spaces, which form the surface of the grate. The grids are interlocked, and

Fig. 59.—Firebar in "Triumph" Stoker

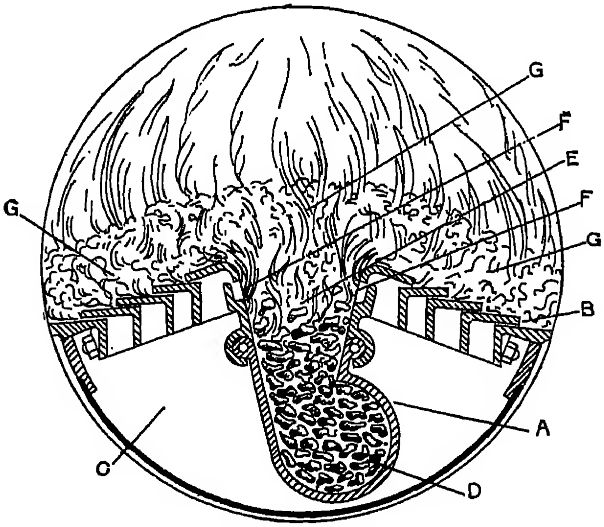


Fig. 60.—Principle of Underfeed Stoker

are locked at the back by a renewable solid block. Scraper rods worked from the front end pass along the bottom of the trough. The steam jets force the air supply along the interior of the trough, where it is distributed through the air spaces in the grids. All wearing parts of the trough are renewable.

Underfeed Stoker.—The principle of the underfeed stoker as applied to the furnace tube of a Lancashire boiler is shown in fig. 60. A retort, or fuel magazine, A, is fitted along the bottom of the furnace tube. In the lower portion of the magazine a taper feeding worm conveys the coal along, and gradually pushes it upwards on to the terraced grates B. As the coal gradually rises it commences to burn at the point E, where incoming air is supplied through the tuyeres F from the wind box C. The coal is thus first coked, the

volatile hydrocarbons mixed with air from the air inlets of the fuel magazine rising through the incandescent mass above, where they are more or less completely burnt, combustion of the solid fuel being completed by the introduction of air through the apertures in the sides of the terraced grates B. By this means the makers claim that as combustion is practically complete by the time the gases have passed through the superincumbent layer of incandescent coal, a clear, short flaming fire results, and the cooling effect of the plates of the boiler does not prejudice the perfect burning of the pro-

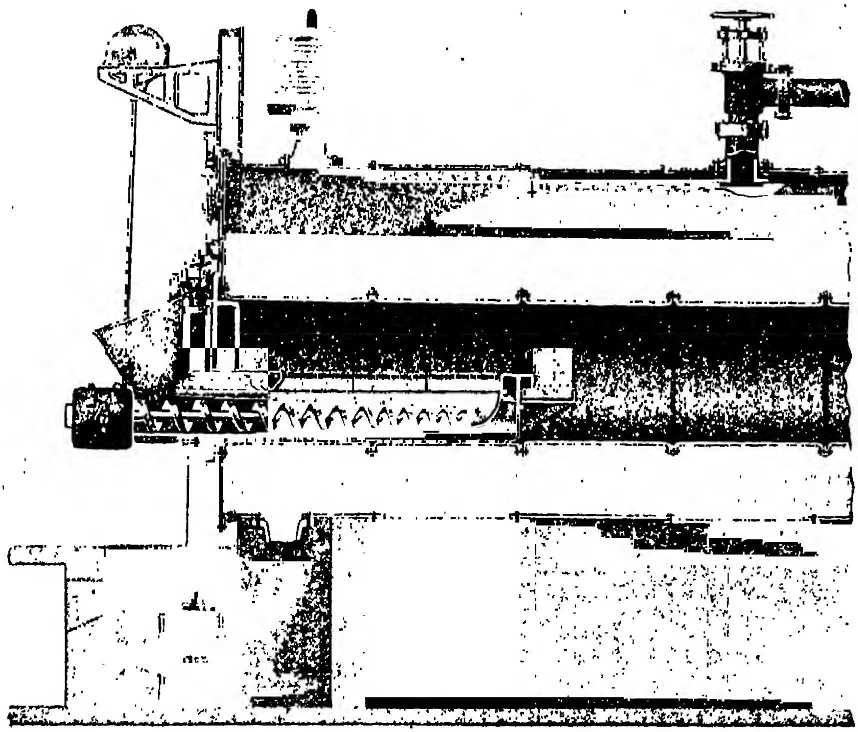


Fig. 61.—Application of Underfeed Stoker to Lancashire Boiler

ducts of combustion. This arrangement, which ensures surface combustion, is generally acknowledged to produce a good fire, but there is a liability to jamming with the worm feed.

A number of different sizes and designs are in existence. The application to an internally fired boiler is illustrated in fig. 61, which shows a stoker manufactured by the Underfeed Stoker Company. The stoker consists of a hopper situated at the front of the boiler, through which the fuel is fed into the combustion retort which runs along the bottom of the furnace tube. A worm conveyor feeds the coal along this retort, at the same time pushing it upwards as already explained. This worm is driven from an eccentric on the main shaft through a ratchet wheel and pawl. The feed may be varied

by the adjustment of a shield at the side of the ratchet wheel, which determines the number of teeth fed by the pawl for each stroke of the eccentric. The air chamber is supplied with air through a cast-iron box situated at the back of the hopper, the air supply being regulated by means of a blast gate. The stoker can be fitted with a double-walled front having air-jacketed fire doors. These jackets receive air from the air box and discharge it in a heated condition over the fire. In order to prevent injury to the mechanism in the event of a jam in the driving worm, a hook-shaped piece of cast iron is inserted between the ratchet wheel on the driving shaft, and the shaft which

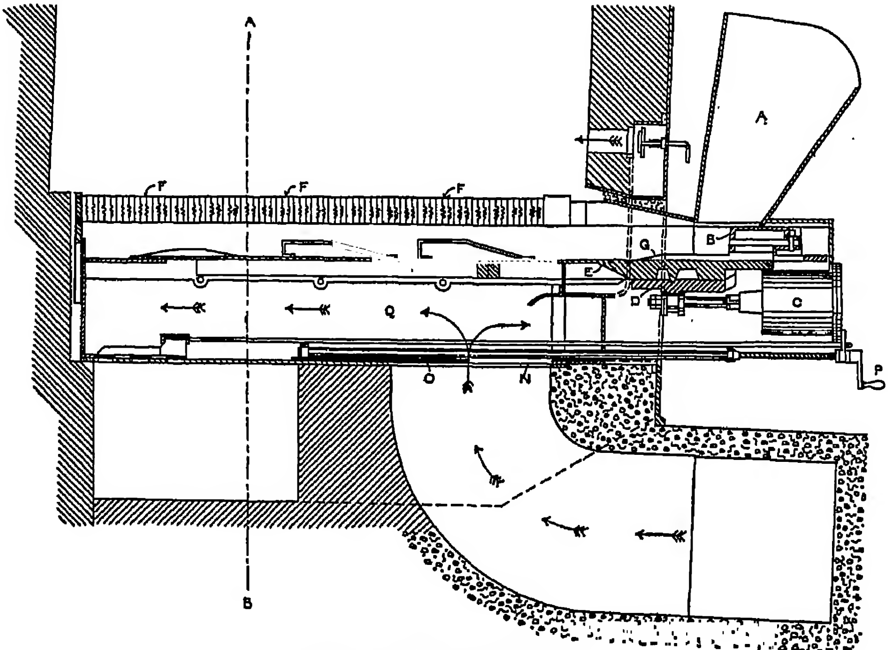


Fig. 62.—Class E Underfeed Stoker

it drives. This breaking link can be inexpensively and easily renewed without interruption to the boiler. The hoppers are so constructed that they may be revolved round out of the way of the furnace doors, thus giving access to the furnace.

The general arrangement of the underfeed stoker for large externally fired boilers is indicated in figs. 62 and 63. Instead of a worm feed, a reciprocating sliding bottom E runs the full length of the fuel magazine or trough. The motion is obtained from a steam cylinder C, the piston of which is connected to the sliding bottom by a crosshead D. A block B, having the same motion as D and E, feeds the coal from the hopper on to the sliding bottom, which gradually carries it to the back of the furnace and at the same time forces it to rise and flood over the firebars F. These bars are alternately moving and fixed, the moving bars working transversely to the retort. The moving bars obtain their motion from longitudinal rocking bars, which

are actuated by two spirals and nuts bolted to the crosshead D outside the furnace. This movement of the grates gradually carries the burning fuel to the sides of the furnace, where the clinker is deposited on a dump plate K, from which it can be dropped into the ashpit by a lever external to the furnace. The arrangement of the air supply is ingenious. The air from a fan enters the aperture N, controlled by the adjustable gate O, into the wind box Q, and, passing upwards, is discharged partly through the holes R into the retort, and partly through the hollow firebars F. This air is discharged at S into

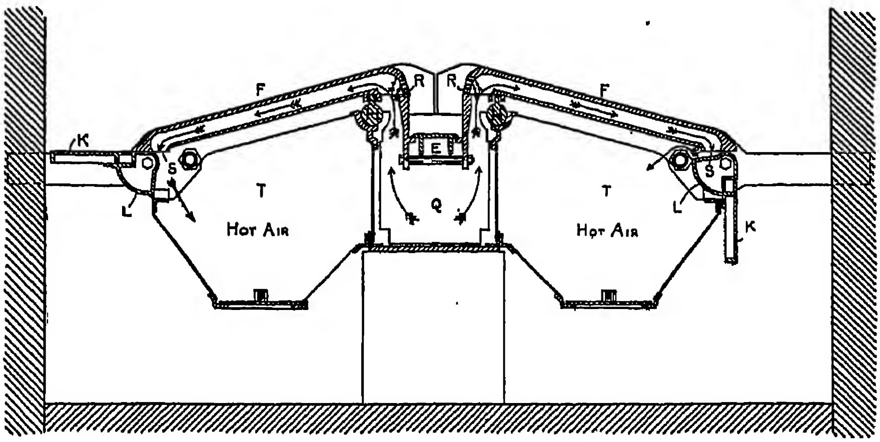


Fig. 63.—Class E Underfeed Stoker

the hot-air chamber T, whence it rises through the spaces between the bars into the incandescent fuel. This action of the air passing through the bars not only obviates overheating of the bars, but also raises the temperature of the air before it passes into the furnace.

CHAPTER VI

Feed-water Heaters

The practice of heating the feed water before injection into the boiler offers a number of advantages. The introduction of cold water into the boiler with the consequent cooling will impose stresses on the boiler due to expansion and contraction set up by the varying temperatures. A high feed temperature will obviate this by producing a more uniform temperature in the boiler, a specially important matter in the case of boilers of the Lancashire and Cornish types.

If a cold feed is employed, heat must be used to raise the temperature of the feed water to the temperature of evaporation corresponding to the working pressure in the boiler. The higher the temperature of the feed, the less is

the heat required for this purpose. In the ideal case, where the feed is at the evaporation temperature, all the heat produced in the boiler is used in actual evaporation of the water. As the higher temperature of the feed water means a quicker change into steam, the circulation is improved. For both these reasons the raising of the feed temperatures ensures a higher rate

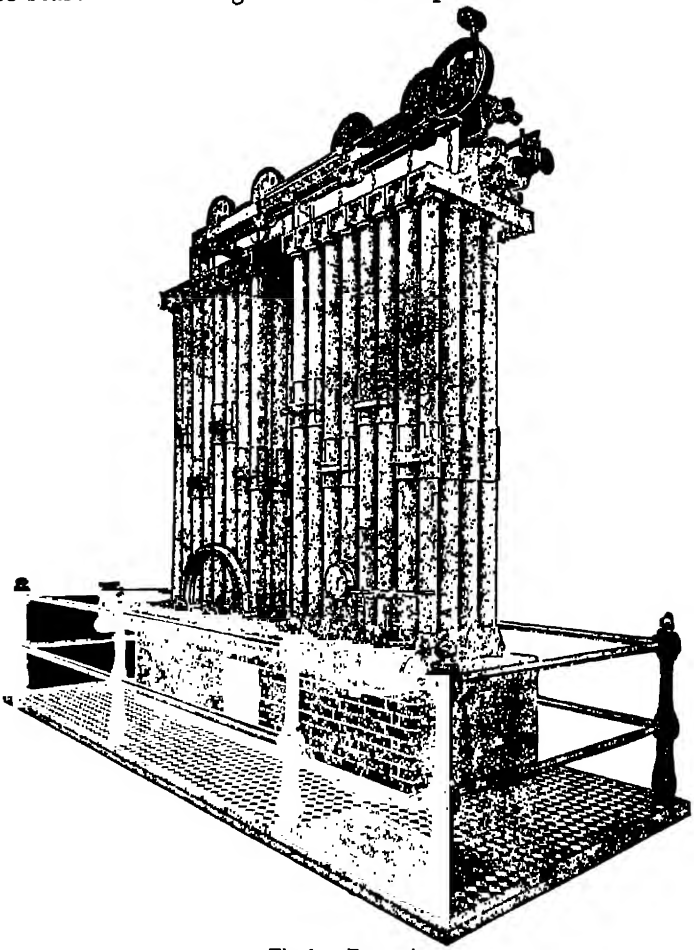


Fig. 64.—Economizer

of evaporation. The heating of the feed water can be carried out by the waste furnace gases, exhaust steam from the engines, or by means of live steam. Apart from the advantages of having hot feed water, the use of waste gases or exhaust steam provides direct economy by the use of otherwise waste heat. With exhaust steam, as the temperature of the exhaust from condensing engines will be comparatively low, it is chiefly of use with non-condensing engines, with which the feed can be heated to about 180° F. In the Weir system of feed heating a portion of the steam from the low-pressure receiver of a compound engine is used to heat the feed water. If the steam

is taken direct from the boiler, the loss due to the fact that the steam might have done work in the engine is theoretically just compensated for by the gain of heat by the feed. If it were taken from the exhaust the heat obtained

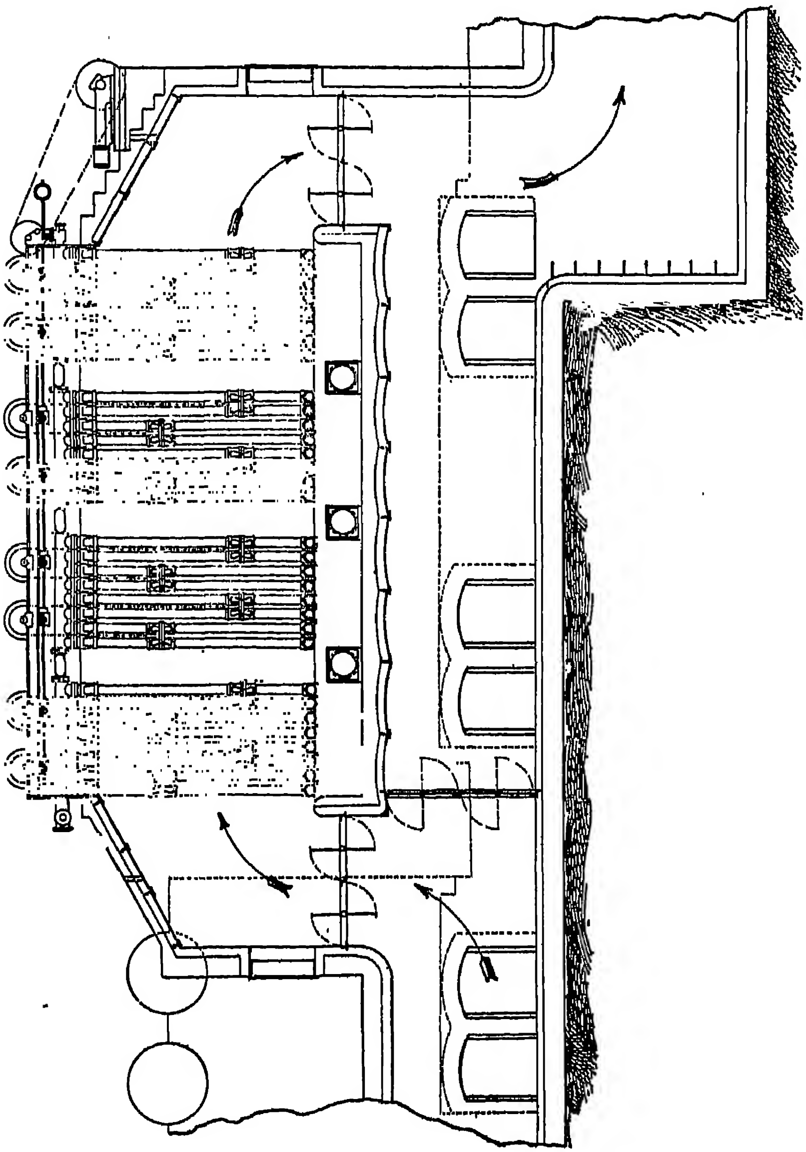


Fig. 65.—Section of Economizer installed at New Avonmouth Docks

is a pure gain, but the temperature of the feed is limited by the temperature of the exhaust. Therefore, by taking the steam from the low-pressure receiver there will be a certain gain in heat and, at the same time, the temperature will be higher.

The Economizer.—This type of feed-water heater, first invented by

Mr. Edward Green, employs the waste heat from the boiler, and is situated in the path of the flue gases before they pass to the chimney. Although in a properly designed boiler the temperature of the flue gases on exit is made as low as possible, yet they carry off a considerable percentage of the calorific value of the fuel. These gases cannot be economically lowered in the boiler proper, as in order to obtain a reasonable rate of transfer of heat they must always be considerably hotter than the boiler temperature. By means of a supplementary heating surface working between lower limits of temperature, the heat which is no longer of service in the boiler can be utilized in raising the temperature of the comparatively cold feed water to something approaching that of the boiler, thus producing a direct increase in economy. As soot, being a bad conductor of heat, will cause a reduction in efficiency, means must be employed to keep the parts of the economizer in contact with the flue gases clean. If the feed water is very cold the condensation on the parts of the economizer where the water enters will cause corrosion to some extent. As the economizer, besides cooling the gases, is more or less of an obstruction in the flue, precautions must be taken to ensure sufficient natural draught, unless some mechanical means is in use.

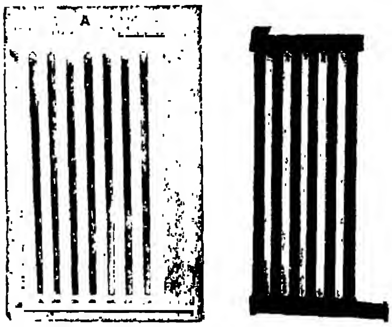


Fig. 66.—Top Section of Economizer Tubes

The general arrangement of such an economizer, made by the Clay Cross Company, is shown in figs. 64 and 65. It consists of a series of tubes fixed top and bottom into headers placed in a flue chamber between the boiler and the chimney. The water, which is admitted at the bottom, passes up the tubes, which are heated by the flue gases which pass between them. The cast-iron tubes are 9 ft. long and $4\frac{9}{16}$ in. diameter. They are forced into the headers in sections of four to ten pipes (fig. 66). The top header (fig. 67) is provided with handholes in which are fitted lids of conical shape. These make a metal-to-metal joint, and are drawn up into position by means of a T-bolt and bridge, being held in position when in use by friction and the internal pressure. The top header is also fitted with a master lid, through the seating of which the other internal lids may be removed, the master lid also being removable through its own seating. The sections of tubes are arranged in groups by top and bottom branch pipes which connect the headers, the bottom branch pipes having access lids, each having four bolts for flushing out purposes. The top headers, being machined on the outside

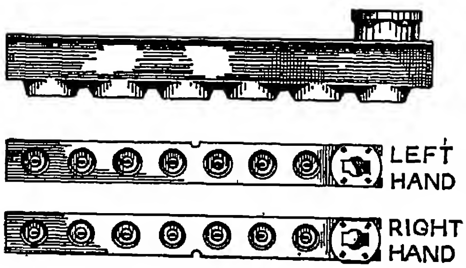


Fig. 67.—Top Header of Economizer

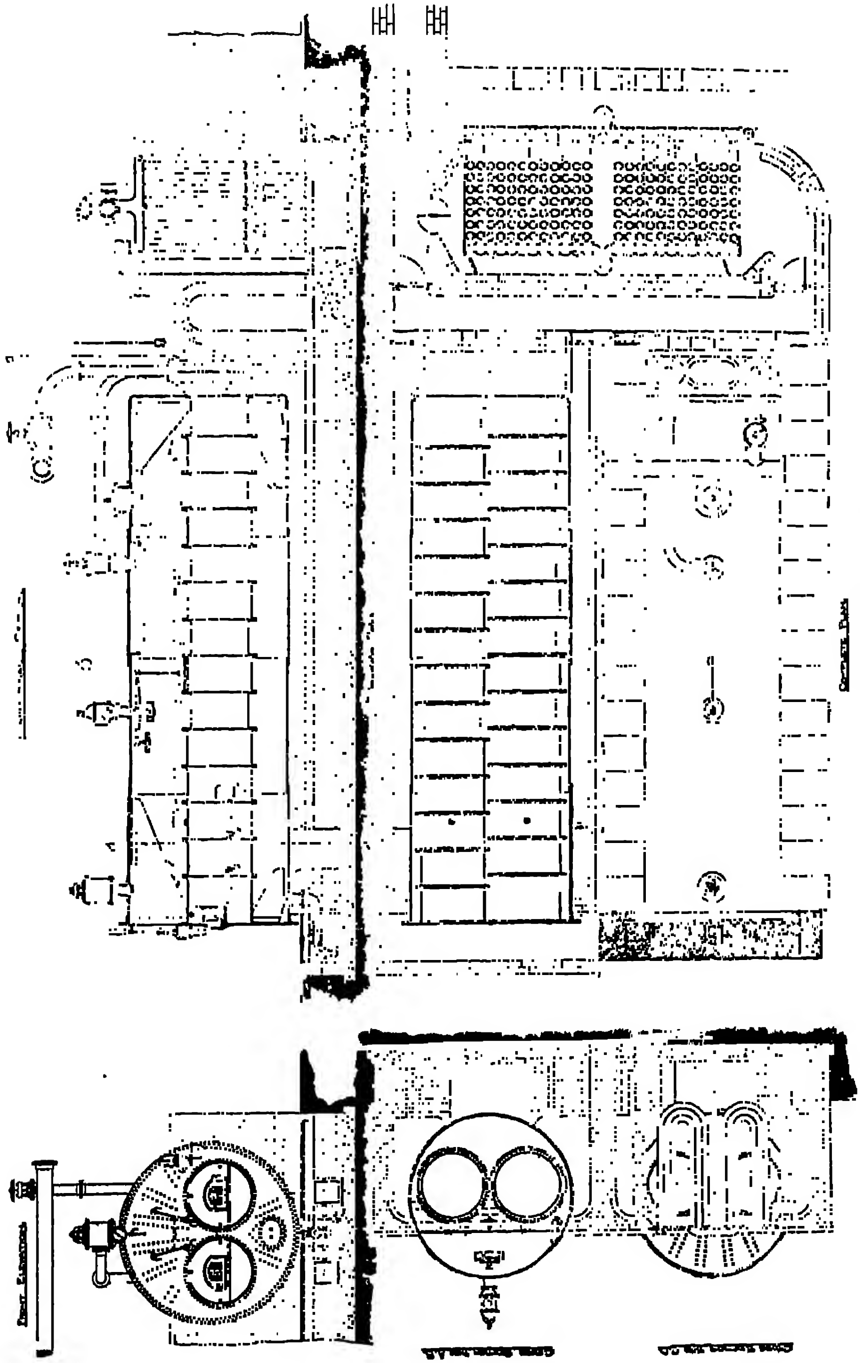


Fig. 68.—Complete Boiler Plant with Economizer

surfaces, fit close together, forming a roof which can easily be made gas- and air-tight. The pipes are kept clean on the outside by means of scrapers actuated by chains and overhead gearing which can be driven by a small steam-engine or electric motor. By means of a reversing gear through which the motion is transmitted, the necessary reversal of the gearing is effected. The economizer is fitted with safety and blow-off valves. Soot doors are arranged in suitable positions for cleaning purposes, while by means of dampers the flue gases can be bypassed without passing through the economizer if desired.

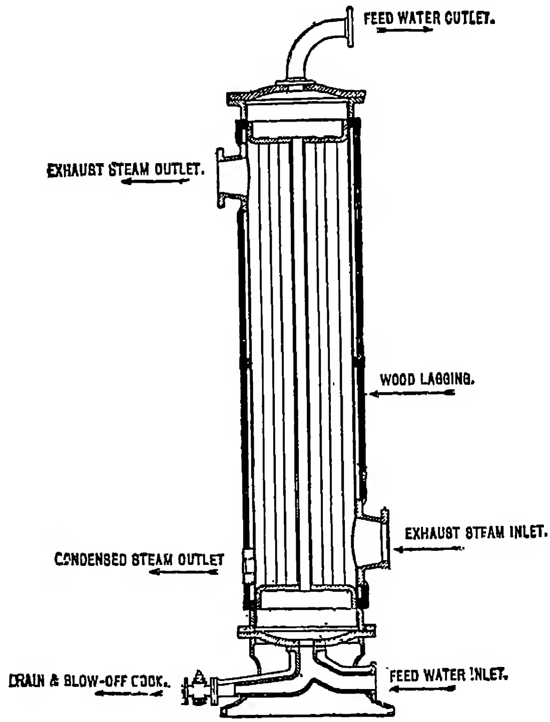


Fig. 69.—Feed-water Heater

The number of pipes and sections used depends on particular conditions, such as the temperature of the flue gases on leaving the boiler, the amount of draught, the rate of evaporation of the boiler plant. To obtain the most satisfactory results care should be taken that there is no air leakage, and that the flow of water should be as uniform as possible. A judicious control of the dampers will have considerable effect on the economy of the whole boiler plant.

A complete lay-out for a modern steam-raising plant with Lancashire boilers, made by Messrs. Daniel Adamson & Co., is shown in fig. 68. It will be noticed that by means of the dampers the economizer can be put out of action if necessary.

Marshall Feed-water Heater.—This type of feed heater, made by Messrs. Marshall & Sons, is useful where it is undesirable to instal an economizer. It can also be used in conjunction with a fuel economizer for taking the chill off the feed water and preventing sweating of the economizer pipes. It consists (fig. 69) of a wrought-steel barrel containing a series of internal brass tubes through which the feed water is passed on its way to the boiler. The exhaust steam is admitted at the lower end of the barrel, and passing round the tubes gives up its heat to the water. End covers are fitted to the barrel to give access to the interior for cleaning, and a blow-off cock is fitted at the bottom through which any deposit may be blown.

Weir Direct-contact Feed-water Heater.—This heater is shown in fig. 70. Steam from the low-pressure receiver of the main engine, and

the exhaust of any auxiliary plant, passes into the heater through the non-return valve B. The feed water is delivered into the heater through the spring loaded valve D. Uniform mixing of the steam and water is obtained by the perforated cylindrical piece forming an annular steam space, and by the perforated disc at the top of this cylinder, through which the water must

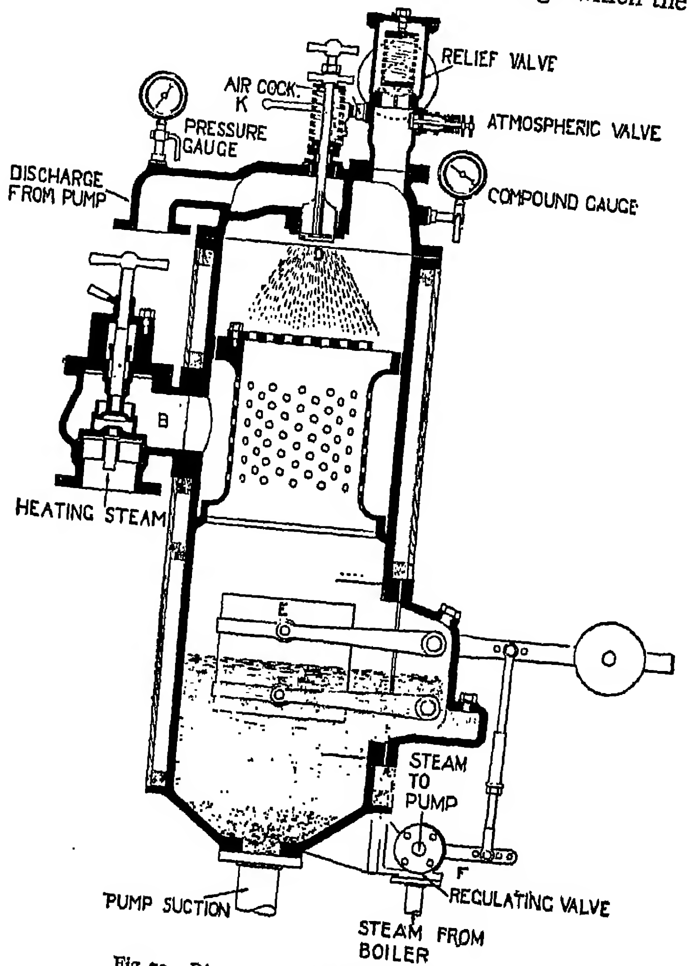


Fig. 70.—Direct-contact Feed-water Heater

pass. As the pressure in the heater is generally less than that of the entering water, any air in the water will be liberated and is removed by means of a cock in the air vessel situated on the top of the heater. This removal of the dissolved air will render the feed water less corrosive. The automatic regulating gear consists of a float E open at the top and suspended on two levers. The top lever is balanced by a weight, and is connected by a rod to another lever actuating the regulating valve which controls the supply of steam to the pump drawing from the heater. The float is always full of

water, and the weight is adjusted to balance when one-half is immersed. The speed of the pump is thus regulated by the quantity of water passing through the heater. A relief valve and two pressure gauges are fitted to the cover, the compound gauge showing the pressure in the heater, while the other shows the pressure of the incoming water.

A method of arranging an economizer in a water-tube boiler is shown in figs. 41 and 77. The feed water circulates through a series of tubes which are heated by the waste gases before they enter the uptake.

CHAPTER VII

Oil Firing

There are many advantages in the use of liquid fuel for steam raising. In the first place, the calorific value is some 30 per cent higher than good steam coal, and the space occupied for equal weights is about 10 per cent less. The storage necessary is therefore considerably less, and it can be stored in places inaccessible for coal storage, an especially important consideration in marine work. The taking in of fresh supplies is a far simpler, quicker, and more cleanly matter than is the case with coal. In addition to this, oil fuel preserves its quality in storage. Fires can be readily started and stopped, and regulation from a low to an intense heat is quickly performed, while the fuel being automatically fed to the burners, no stoking or trimming is necessary, and no ashes or clinkers have to be dealt with. Full steam pressure can therefore be readily obtained, and a steady head of steam continuously maintained without waste, and there are no "stand-by" losses. Stoking, trimming, and cleaning being done away with, there is considerable saving in labour. As the supply of air can be easily regulated, and there are no furnace doors to be opened, more perfect combustion with a more equal distribution of heat is obtained. This means cleaner heating surfaces in the boiler, less heat lost in the uptake, and a consequent increase in efficiency, with a cleaner and cooler boiler-room. With a well designed furnace, and ordinary care, smoke can be eliminated. Owing to the particular characteristics of oil fuel, certain precautions must be taken in its use. Owing to the comparatively low flash-point, all pipe lines should be kept free from leakage, and the question of storage is a matter for careful consideration, especially in congested districts. With reasonable care in its use, oil fuel should not entail any special risks.

To ensure complete combustion and intimate contact with the proper amount of air, the oil must be atomized in a burner, this mixture of oil spray and air being burned in the furnace. One of the most important things in the combustion of oil is the question of the correct supply of air. This is generally obtained by means of a hand-regulated damper. A preliminary

heating of the oil within the flash-point temperature will increase the economy. In order that the burners should not become clogged, liquid fuel must be carefully strained before passing to the furnace.

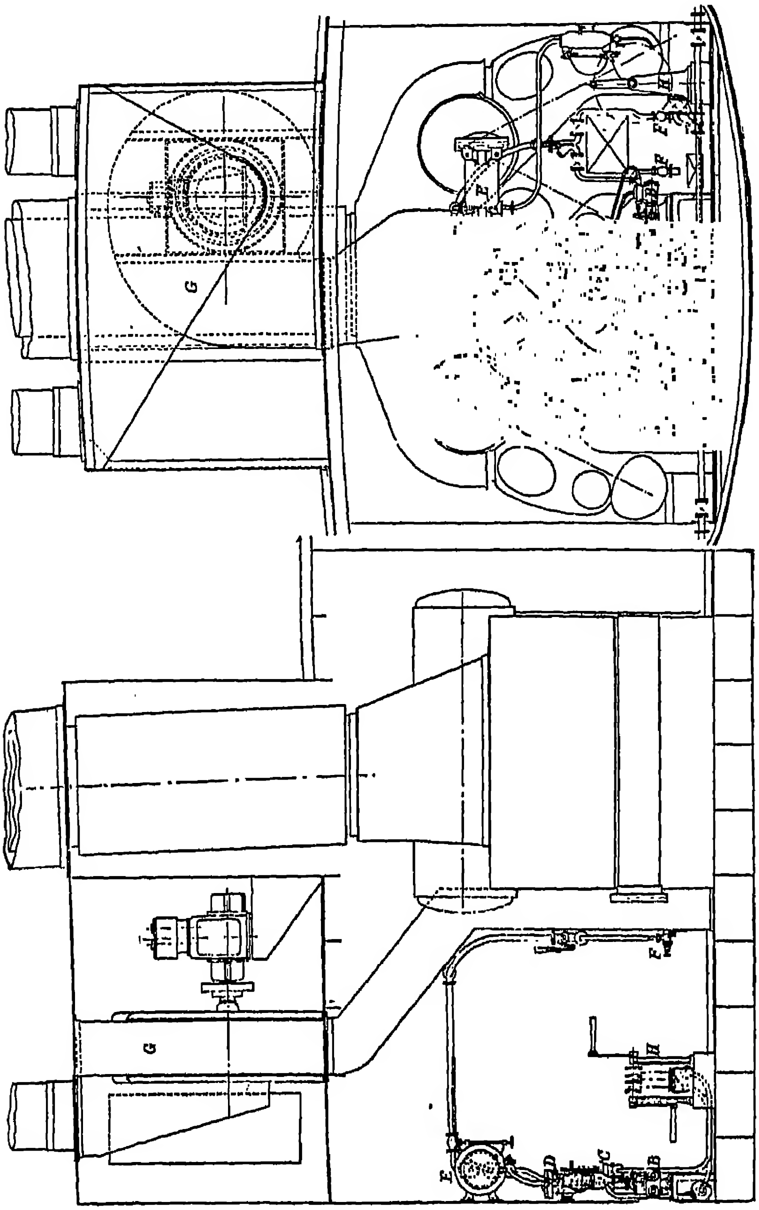


Fig. 71.—Pressure-jet System of Oil Firing

There are three systems in general use for atomizing the oil. In the pressure system, the oil, after heating and straining, is forced under pressure through a specially constructed burner. This method, which is compara-

tively simple in operation and on the whole most efficient and convenient, is the one generally used in marine practice. In the air-jet system, the fuel is atomized by means of air at a small pressure. Although a certain amount of steam is required to run the compressors, this may be returned to the boiler in the usual way. This method can be used not only with crude petroleum, but also with alcohol, creosote, blast-furnace oil, &c. The steam-jet system, in which steam is used to atomize the fuel, is largely used for factory boilers of all kinds. This system can be fitted to boilers without alteration to the furnace, so that coal or oil can be used. Where only oil is to be used, however, it is better to remove the firebars and arrange the furnace specially. This system has the advantage of occupying a small space, and

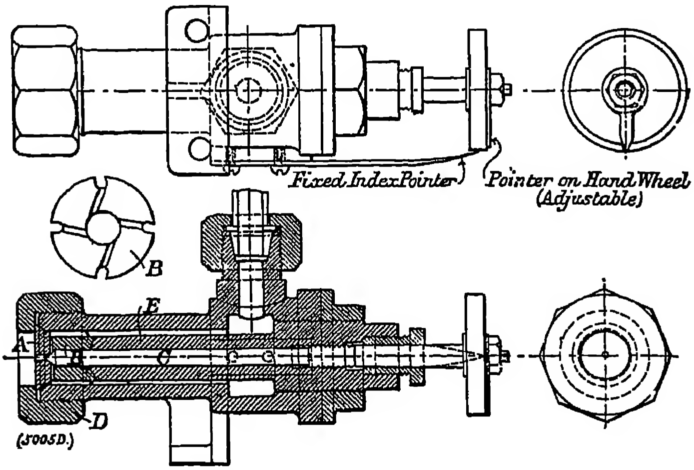


Fig. 72.—Pressure-jet System Burner

the atomizing agent is available in unlimited quantities as long as one boiler is working. When starting up, a supply of steam is required, although in certain locomotive running sheds compressed air is sometimes used. About 4 per cent of the steam raised is required for atomizing the fuel, a serious matter in some cases, especially in marine practice.

Kermode's Pressure-jet System.—By this method it is claimed that 80 to 83 per cent of the calorific value of the fuel used is recovered in actual work. A typical installation on this system, as fitted to two Yarrow-type boilers, is shown in fig. 71. The boilers are arranged for forced draught upon the open stokehold system. The oil is drawn from the tanks through the pipes AA by means of the steam-driven oil-fuel pumps BB. It is then delivered through the first set of duplex oil straining filters DD to the steam oil-fuel heater E. The heated oil is then supplied through the second set of straining filters DD to the burners FF. A hand pump H is provided for initially raising steam. When starting from cold water, it may be necessary to warm the oil. This may be performed in several ways. The general method is by means of a special burner combined with a vaporizer, in which the fuel is

heated by the flame impinging on it. A second method is by heating the oil fuel in a special tank by means of a blow-lamp, the oil then being delivered by the hand pump to the whole system. A third method uses a small boiler to produce steam for heating the fuel heater E, the oil fuel being circulated by means of the hand pump until the correct temperature is obtained. The usual working pressure of the oil is 150 lb. per square inch, and the temperature of the oil at the burners about 200° F.

The burner is shown in fig. 72. The oil fuel enters the burner at the side and passes along the annular space between the outer wall and the inner cylinder E. This cylinder is screwed into the shell of the burner and abuts at its front end against the cap nut A. At the front end of the cylinder E is a detachable piece B, which for a part of its length is enlarged to fit the interior of the outer shell. This piece B is grooved on its front face with the grooves tangential to the central hole in it. Through these grooves the oil obtains access to the discharge opening, which is controlled by the conical end of the spindle C, the position of which can be adjusted by a handwheel fitted with an adjustable pointer and a fixed index pointer. The burners are so made that they are controllable either by varying the oil pressure or by adjusting the valve so that the oil discharge can be varied from a few gallons to 100 gallons or more per hour.

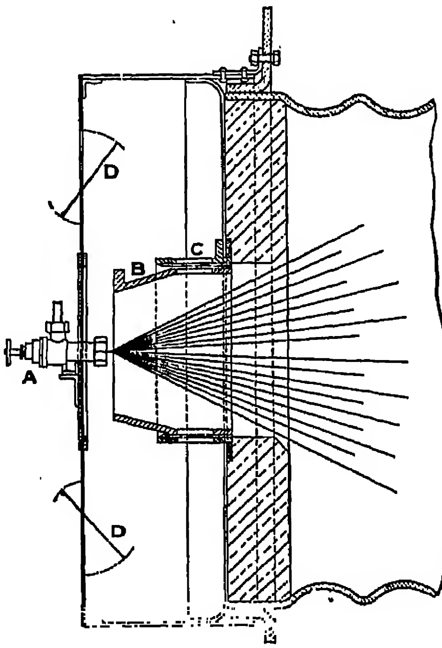


Fig. 73.—Air Control

The method of air control is shown in fig. 73. A casing to which the burner A is fixed is fitted with two automatic louvres D. These louvres will swing open to an adjustable amount to allow air to pass into the furnace, but will automatically close in the event of a back-fire in the furnace. The burner sprays through the air cone, which consists of an inner and outer tube, B and C. Air passes through the ports C and can be adjusted to the correct amount by adjusting the inner tube B, which can be " tuned " from the stokehold. The ports C are inclined to the radial direction in order to give a whirling motion to the air.

The following table gives a summary of results of tests with Kermode's pressure-jet system with natural and forced draught. In trials Nos. 2 and 3, firebars and doors were in position.

KERMODE'S PATENT PRESSURE-JET SYSTEM, WITH NATURAL AND FORCED DRAUGHT

	No. 1	No. 2.	No. 3.
Type of boiler	Cornish	Yarrow, Naval	Yarrow, Naval
Class of oil used	Mex. Fuel Oil	Texas Fuel Oil	Texas Fuel Oil
Specific gravity of oil	0·947	0·955	0·955
Average feed temperature, degrees F.	45	44	44
Steam pressure, pounds per square inch	65	250	250
Description of draught	Natural	Forced	Forced
Air pressure, inches of water	0·25	0·61	2·25
Description of smoke	Slight	Trace	Trace
Evaporation per pound of oil actual ..	13·9	12·98	11·15
EQUIVALENT EVAPORATION:—			
From and at 212° F. per pound fuel	16·74	16·0	13·77

Kermode's Air-jet System.—In this system the oil is sprayed by means of air at a pressure of from ½ lb. to 4 lb. per square inch, according

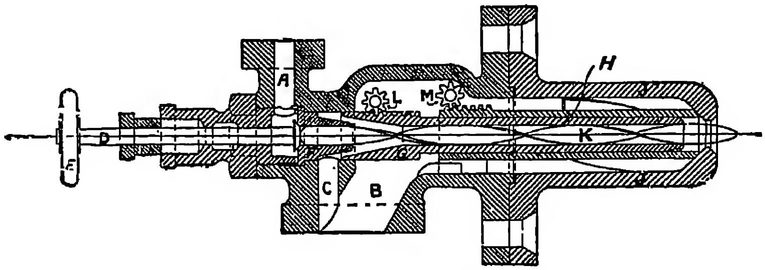


Fig. 74.—Air-jet System Burner

to the particular conditions of the boiler plant. It is claimed that 83 per cent of the calorific value of the fuel used is recovered in actual work, less than 2 per cent of the steam raised being used to drive the air-compressing plant. The air compressor can also be driven electrically, or from an existing power shafting.

The oil enters the burner at A (fig. 74), its supply being controlled by a valve actuated by the handwheel E. Heated compressed air enters at C and, mixing with the oil, sweeps it along the centre passage in the burner. This current of air can be regulated by moving the cylinder G by means of the rack and pinion L. The mixture of oil and air is given a whirling motion by the spiral K, which is fixed on the front of the spindle D. This assists the vaporizing of the oil and the mixing with the air. A second supply of air passes from B into the annular space between the outer casing J and an inner tube. This air, which is given a whirling motion by spiral guides in the annular space, meets the oil-and-air mixture at the front of the burner just where combustion commences. The supply of air is regulated by shifting the cylindrical tubes by the rack and pinion M. A third supply of air is induced outside the burner by the draught in the furnace. This

burner, which can be used for all kinds of liquid fuel, will deal with mixtures of tar and creosote, or pitch and creosote.

Kermode's Steam-jet System.—The burner in this system is shown in fig. 75. The oil enters by the pipe B and passes into the hollow cone H, the supply being regulated by a valve on the spindle A, which is regulated by the handwheel N. The oil has a whirling motion imparted to it by the spiral stem G on the front of the oil-valve spindle. The steam enters by the pipe C and, passing round the hollow cone H, forms a steam jacket to the oil. The air passes into the air cone P through the ports D, by the inductive action of the steam. This air is given a whirling motion by spiral guides. The oil and steam mix with the air at P, the whole mixture being ignited and burning with a large flame. The amount of steam issuing at the opening between the two cones F and H is regulated by rotating the air cone P which moves the hollow cone F with it. The air supply is controlled by a perforated

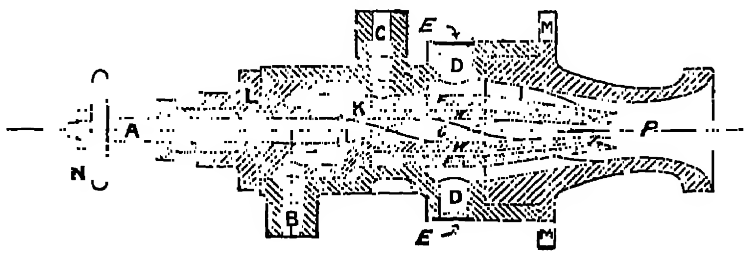


Fig. 75.—Steam-jet System Burner

ring E which varies the size of the openings of the ports D. This system, which can be used on all types of boiler, will handle crude petroleum tar, creosote, water-gas tar, Mond-gas tar, mixtures of pitch and creosote, and various vegetable oils. It is claimed that the steam jet will recover from 70 to 75 per cent of the calorific value of the fuel used.

The application of this system to Lancashire boilers is shown in fig. 76, where both oil and coal firing can be used. In the case of oil firing, the firebars are covered by a layer of broken firebrick. The oil tank is arranged above the boilers, the necessary steam being supplied to the burners by branch pipes. Dampers are fixed in the front of the furnace below the line of the firebars to control the supply of air.

Adoption of Oil Fuel to Boilers.—Although oil fuel is used in the Lancashire and Cornish type of boiler, they are not really suitable to withstand the intense heat generated. For this reason it is generally necessary to line the furnace tubes with some refractory material. This covering up of what is normally the most active part of the heating surface makes a considerable difference in the heating surface area, besides reducing the effective radiation. For this reason, when oil firing is exclusively used in a Lancashire boiler the bottom of the furnace tubes only is lined with firebrick, the oil burner being inclined downwards. Oil fuel can be used with vertical boilers if suitable precautions are taken to protect the boiler

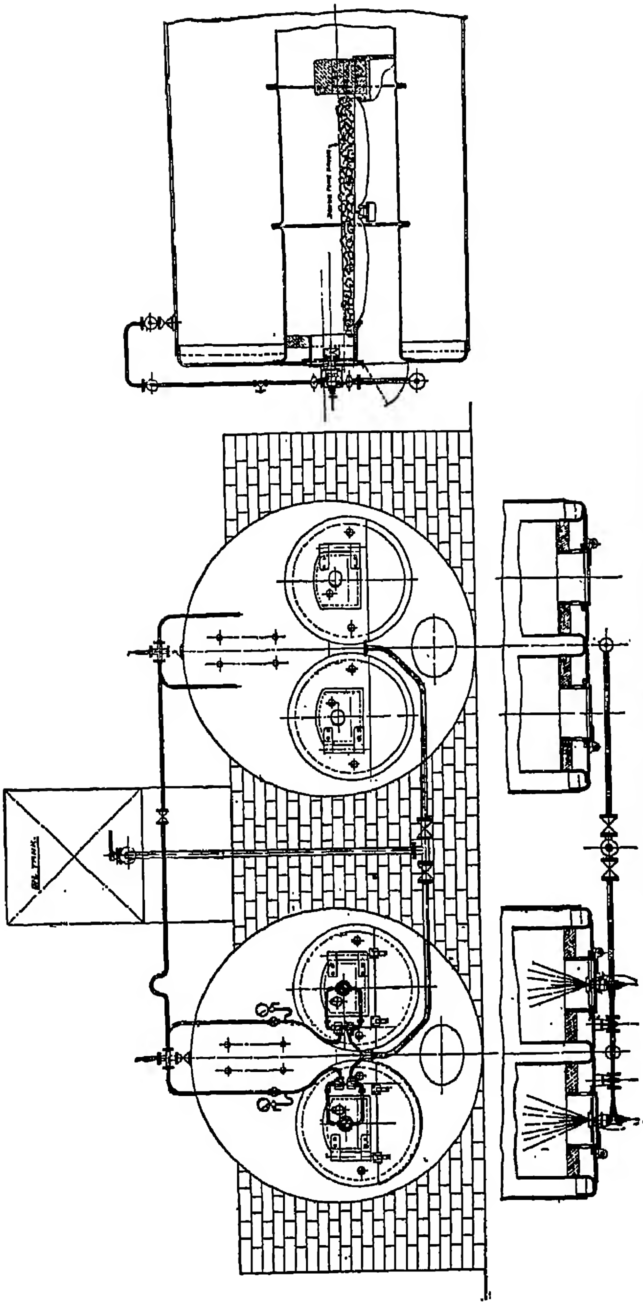


Fig. 76.—Application of Steam-jet System to Lancashire Boilers

plates where the heat is intense. With a vertical boiler of the Cochran type, the capacity of the furnace can be increased by mounting the boiler on a ring of brickwork and introducing the spray beneath the boiler. Care must be taken to cover the bottom seams with brickwork to protect the rivets from

the intense heat. If a brickwork foundation is not permissible, the bottom plate of the boiler may be extended and lined with firebrick.

In the case of the water-tube boiler with its large combustion chamber

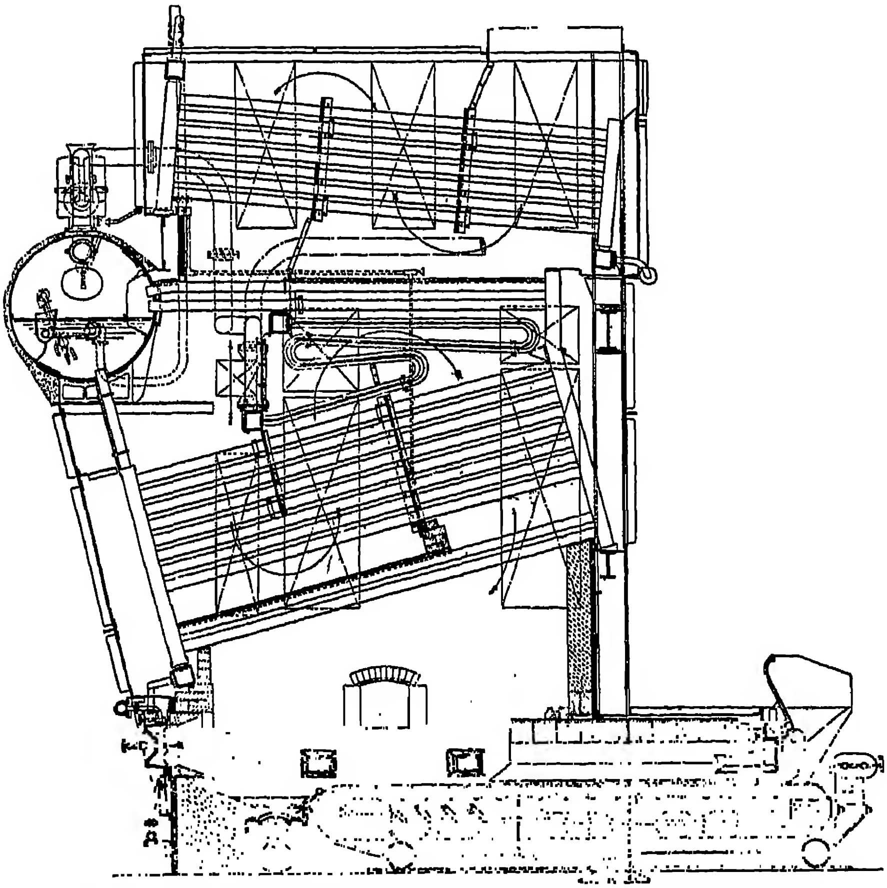


Fig. 77.—Marine-type Boiler (for land purposes) fitted with Superheater, &c., and Oil Firing

lined with brickwork, the problem is much simpler. No part of the boiler proper is in contact with the most intense heat, and the large size of the combustion chamber ensures complete combustion. Fig. 77 shows a Babcock & Wilcox marine-type boiler arranged with oil-fuel apparatus and chain-grate stoker

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